COMBUSTION

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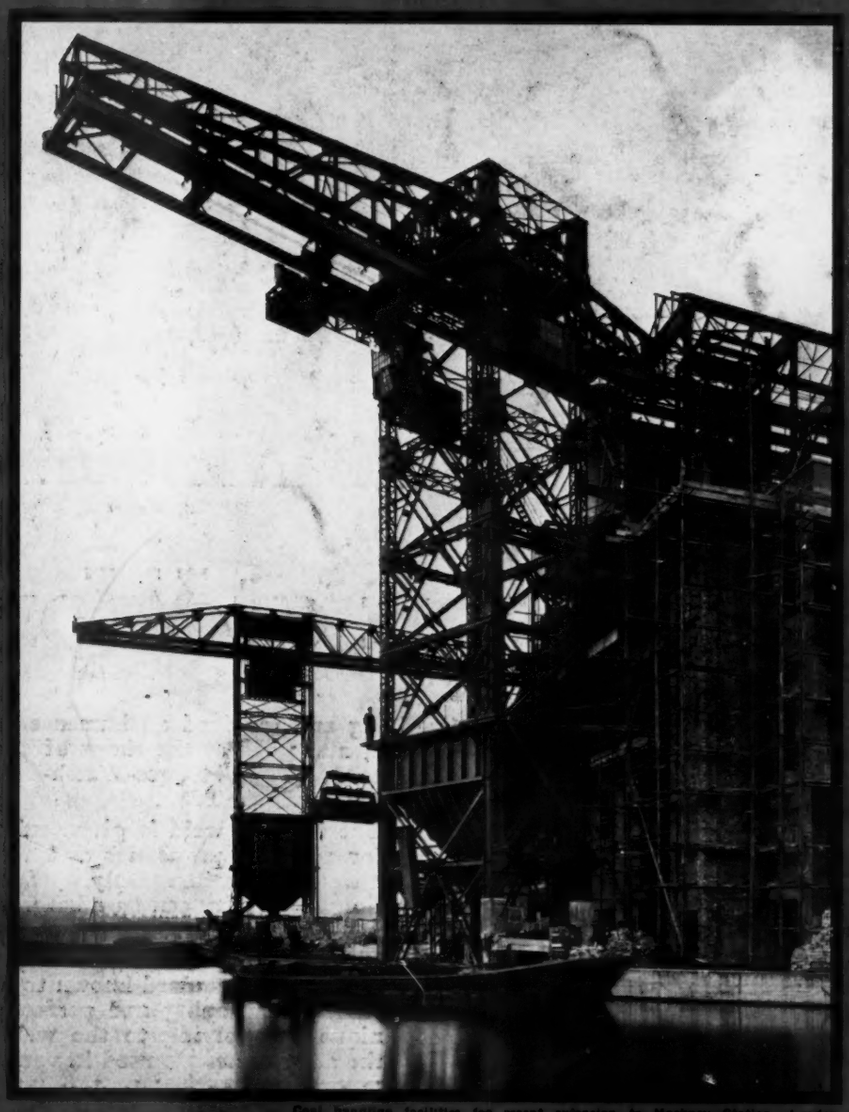


Coal handling facilities for recent extension to Hackney Station, London

Water-Cooled Furnaces for Chain-Grate Stokers

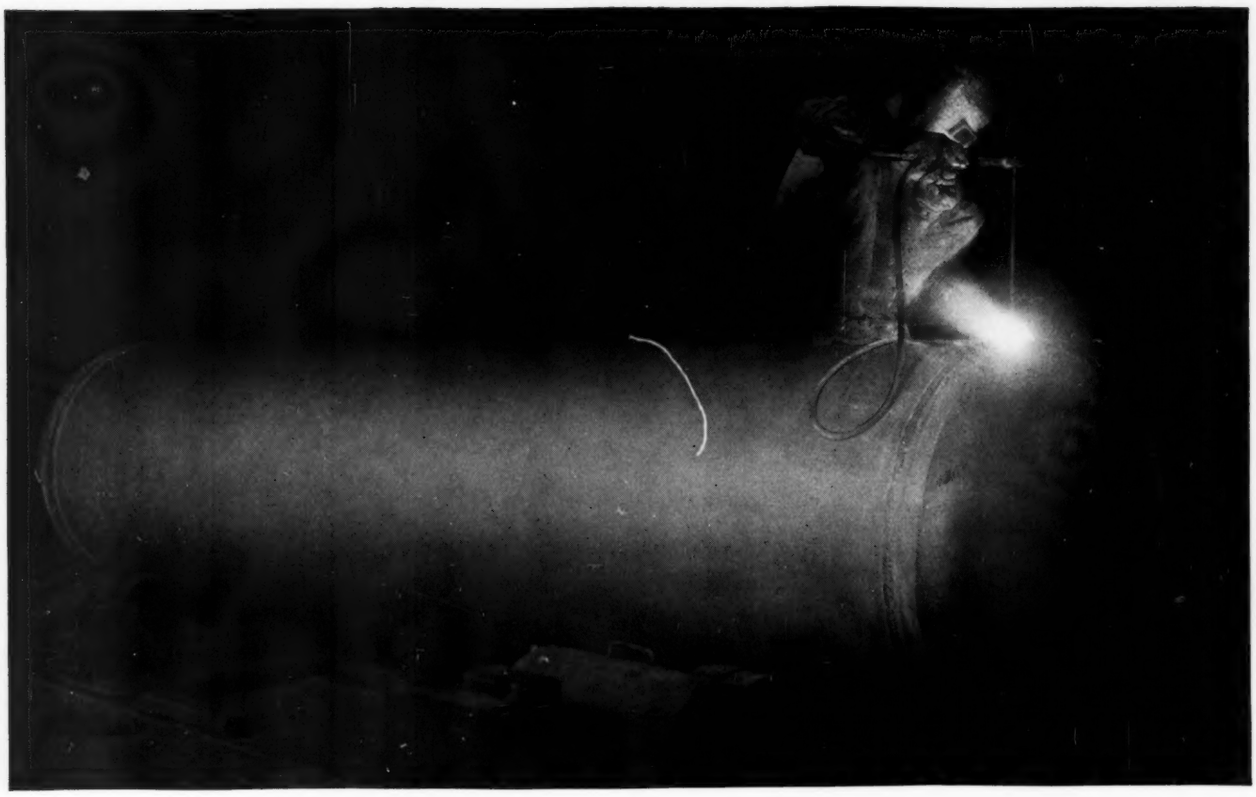
By Otto de Lorenzi

The Performance Characteristics of Centrifugal Boiler Feed Pumps

By J. G. Mingle

OTHER ARTICLES IN THIS ISSUE BY

JOHN T. FARMER • P. B. PLACE • DAVID BROWNLIE



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COMBUSTION

VOLUME FOUR • NUMBER ONE

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$8235.00 → This is the yearly savings in coal made by the installation of a Type E Stoker in a steel company's plant in Ohio. This plant was burning 4900 tons of coal at $3.15. It now burns 3600 tons at $2.00.

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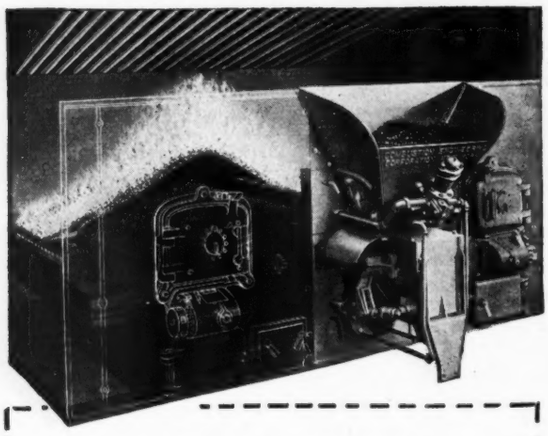

In each of the above installations the quantity of steam being produced by Type E Stoker firing is materially increased. Consequently, the ultimate coal savings are even greater than those indicated depending upon the amount of the additional capacity being utilized. There are still further savings due to reduced maintenance and labor costs.

You will be interested in an article entitled "Modernizing the Old Boiler Plant" in which the savings in two of the cases listed above are explained in detail. A copy will be sent upon receipt of the coupon below.

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No. of boilers in plant..... Total Boiler hp.....

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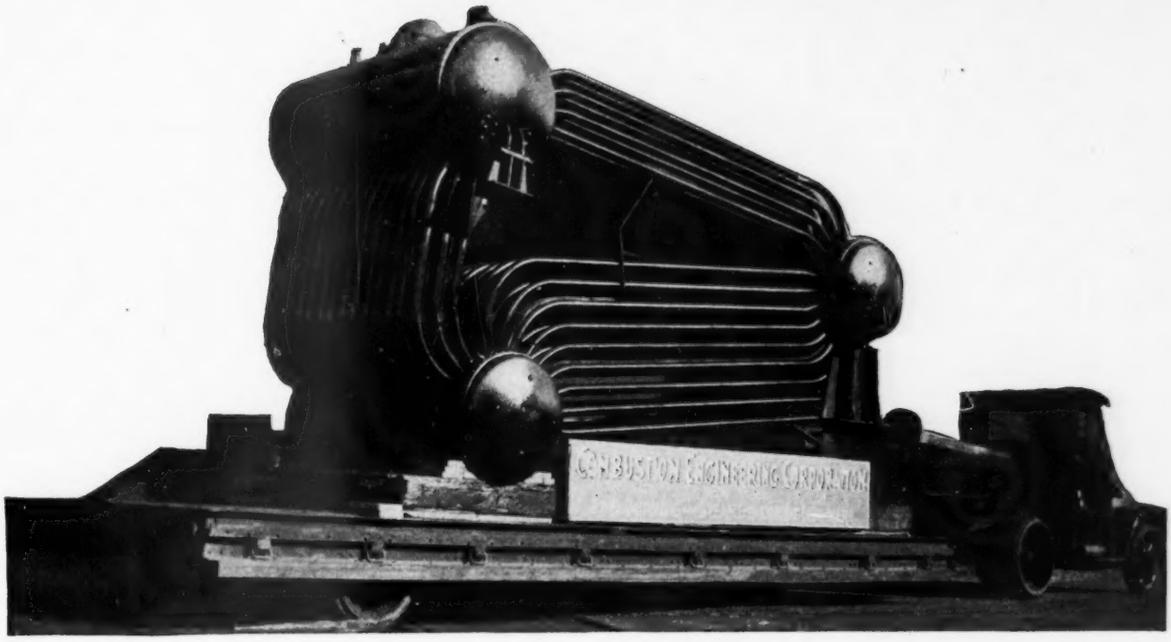

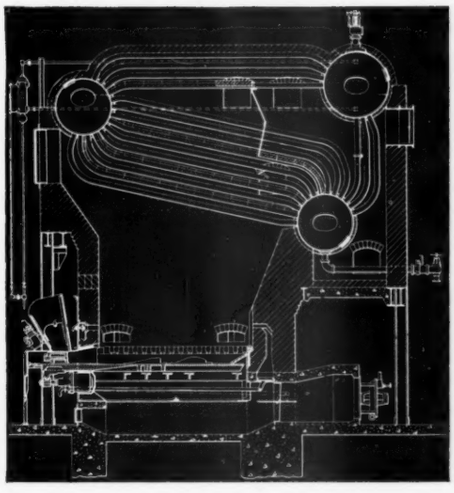

Typical application of C-E VM Type Boiler fired by a Type E Stoker.

GOING TO MARKET

This is one of two completely assembled C-E VM Type Boilers being delivered to the Firmin Desloge Hospital in St. Louis, Missouri. These boilers are of the low head-room type, each containing 2,340 sq.ft. of heating surface and operating at 180 pounds pressure.

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EDITORIAL

A Warning to "Low-Bid" Buyers

IN times of stress sound purchasing policies, like all other business policies, are apt to be departed from, with consequences that eventually are very harmful to the purchaser. The buyer who is overanxious to take advantage of a highly competitive price situation is on dangerous ground. Lured by the bait of what appears to be an extremely attractive price, he is likely to overlook one of the first principles of sound purchasing procedure, namely that the most economical buying is not necessarily buying at the lowest price. Especially is this true with respect to mechanical equipment, where the factors of low maintenance, low operating cost, reliability and efficient performance are far more important from an actual money standpoint than first cost.

In the power equipment field, first cost is usually a substantial item and obviously should be regarded in the light of an investment, a satisfactory return from which can be depended upon. Equally obvious is the fact that it is poor economy to jeopardize this return and incur serious collateral losses, due to unreliable and unsatisfactory performance, in order to save a few dollars in first cost.

There has been too much "low-bid" buying in the equipment field during the past year or two. All too often equipment has been bought at a price which, within the course of a couple of years, will show itself to have been a costly piece of purchasing instead of the shrewd bargain it was thought to be at the time.

There are certain organizations in every branch of manufacturing who have a reputation of many years' standing. These organizations cannot afford to jeopardize their reputations and their futures by cheapening their equipment to a point where it will not give the performance expected of it. Performance is not merely a matter of figures; efficiency and maintenance guarantees are only as good as the manufacturer's word. Specified performance cannot be obtained from a machine unless it is an inherent quality of that machine—the product of its design and the materials of which it is built. This is something which the buyer who is primarily interested in the lowest price is apt to overlook, but the consequences cannot be evaded and will constitute a recurrent indictment of his judgment throughout the life of the equipment.

Of course, it is the buyer's responsibility to buy at the lowest figure consistent with satisfactory results, but unless such a price can be procured from a manufacturer whose experience and past reputation assure the type of service expected from the equipment, the ultimate cost of the purchase is largely a matter of conjecture. Certainly this element has no place in the business of equipment purchasing. It goes without saying that the buyer who takes advantage of bids from unreliable sources in order to exact a profitless price from a responsible manufacturer is not only guilty of unfair practice but is contributing to a condition of competition from which he will ultimately suffer.

Combustion, Vol. IV, No. 1

WITH this issue, *Combustion* begins its fourth year of publication. In our first issue, July, 1929, we summarized our editorial policy as follows:

> "The principal aim of *Combustion* will be realized if it effects a broader understanding of the important developments that are constantly taking place in this field, and thus contributes to the more extended use of better principles and practice in the arts of fuel utilization and steam generation."

We believe we have accomplished this aim not only in the broad sense of the words in which it is stated but in the more specific sense of having published a great many articles which may be characterized as notable contributions to the literature of the subjects to which they pertain. The permanent value of these articles is perhaps best indicated by the frequency with which they appear in both American and foreign bibliographies. As an example of this, one prominent British publication recently listed and summarized in a single issue seven articles which had their original publication in *Combustion*.

Many letters from our readers indicate that in the case of *Combustion* they have abandoned their usual practice of clipping certain articles for their reference files because all the articles have permanent value and consequently they have found it necessary to retain each issue intact.

These splendid evidences of recognition give us cause for much gratification as we pause at the threshold of our fourth year and view our work in retrospect. They have been the principal sources of our encouragement during the three difficult years through which we have passed and have constantly renewed our ambition to make *Combustion* an even better and more valuable magazine. We feel a natural urge to share with our readers these things that have meant so much to us, and to express our very real appreciation of the many encouraging and helpful letters we have received. Needless to say, we hope to have still more of these letters during the coming year and cordially invite the opinions and comments of our readers everywhere.

Water-Cooled Furnaces for Chain-Grate Stokers

By OTTO de LORENZI
Combustion Engineering Corporation
New York

This article complements Mr. de Lorenzi's discussion of water-cooled furnaces for underfeed stokers, published in the April issue. While the primary purpose of the present article is to relate the development of water-cooling for chain-grate stokers of both natural and forced draft types, the author necessarily discusses the collateral development of arch design and arrangement. The latter phase of chain-grate-stoker furnace design is one in which there has been particularly rapid progress in recent years. This progress has been concurrent with the adaptation of the water-cooled furnace to chain-grate stokers, and each development has influenced and contributed to the other. . . . . The latter part of the article discusses present-day practice in furnace and arch design and the use of water-cooled surfaces with relation to both natural and forced draft stokers and to different coals.

SOME form of water-cooled furnace lining has been in use with the chain-grate type of stoker for many years. It is true that the function of this surface, in the earlier installations, was not so much to reduce maintenance costs, as to improve operating efficiencies. However, the gradual development and application of furnace wall cooling for this type of stoker may be said to have evolved from the water-back and the side wall water box. It will be of interest therefore to briefly describe and illustrate these forerunners of the modern types of furnace constructions.

The early chain-grate stokers were of the natural draft type. In the operation of these machines the air for combustion is induced, by furnace suction, through the fuel bed. In order to prevent bypassing it is necessary to fully cover, with fuel, the air-admitting grate surface from grate to discharge end of the stoker. If this active fuel bed is too short or thin and full of holes, a large quantity of air will find its way into the furnace and serve no useful purpose in the combustion process. The result is an increase in the excess air in the gases leaving the furnace, over the actual quantity required to effectively burn the fuel. The net result of this method of operation is an increased heat loss and a corresponding reduction in the operating efficiency. On the other hand, if the active fuel bed is carried to the end of the stoker, a large quantity of unburned carbon will find its way into the ash pit. The heat loss from this method of operation may equal or exceed that existing under the conditions of high excess air, due to a short and spotty fuel bed. To minimize the losses just described it becomes necessary, therefore, to provide a suitable furnace seal and fuel dam. A properly located water-back will function in this twofold manner.

Prior to the use of water-backs the furnace seal, at the rear end of the stoker, was provided by building an overhanging bridge wall. This overhang was constructed of refractory material, and incandescent ash would adhere to it. Clinkers would quickly form and clog up the ash discharge opening. The overhang would soon burn off and become ineffective. By providing a water-cooled support in the form of pipes tied into the boiler circulation, it was found that these operating difficulties could be remedied and a constant opening maintained. The height at which this water-back is set above the grates varies from 3 to 6 inches, its setting being determined by the ash content of the fuel being burned. It should not be higher than necessary, to permit the ash of a normal fuel bed thickness to pass under. For mid-western bituminous coals this distance is generally 4 inches. Aside from forming an air seal and fuel dam, the water-back also serves to compress the fuel bed and therefore increase its density at the rear of the stoker. In this manner thin spots are eliminated and a relatively uniform fuel bed resistance is maintained. As a result excess air requirements are minimized and the combustible loss in the ash pit reduced to a low value.

Another source of excess air as well as combustible loss is the formation of clinkers where the edges of the fuel bed come in contact with the furnace side walls. These clinkers form effective ploughs which force the fuel from the sides of the grate surface. A large percentage of the combustion air will then pass into the furnace through these bared areas. The fuel bed will burn out at the sides while the center will discharge consider-

able quantities of combustible matter into the ash pit. Frequent slicing of the side wall will serve to minimize this loss to some extent. However the formation of these clinkers, the source of the trouble, may be eliminated by the installation of side wall water boxes. These boxes are most effective when tied into the boiler circulation. In some instances the water is supplied at ordinary service pressure. The flow is maintained at a rate which assures an outlet temperature not greater than 175 deg. fahr. The heat absorbed, by this cooling water, is then either wasted or enters the boiler by way of the feedwater heater. The effectiveness of both the water-back and the side wall water box is illustrated by the fact that they will absorb from 1½ to 3 per cent of the heat supplied to the steam generating unit. Because of this capacity for heat absorption, they should invariably be tied into the boiler circulation, and thereby aid in maintaining maximum operating efficiency.

A conventional application of a natural-draft, chain-grate stoker to a bent-tube boiler is shown in Fig. 1. This furnace is designed to burn Illinois coal and develop 200 per cent of boiler rating. Of particular interest will be the design and location of the arch forming the roof of the dutch oven. The front end of this arch is only 3 ft. above the grate surface, while the rear end is somewhat higher. The arch length is 8 ft. 6 in. while the effective stoker length is 11 ft. A side wall water box, tied into the boiler circulation, is provided to prevent the formation of side wall clinkers. This cooling surface extends to a distance of 8 in. above the stoker grate. The water-back, set 4 in. above the grate, is composed of two tubes which are also tied into the boiler circulation. These tubes, in addition to forming the necessary seal and fuel dam, act as a support for the overhanging bridge wall. There are many installations of this general type that have been in operation for more than fifteen years. Their success is due, in a very large measure, to the installation of these simple yet effective means for the prevention of objectionable clinker formations.

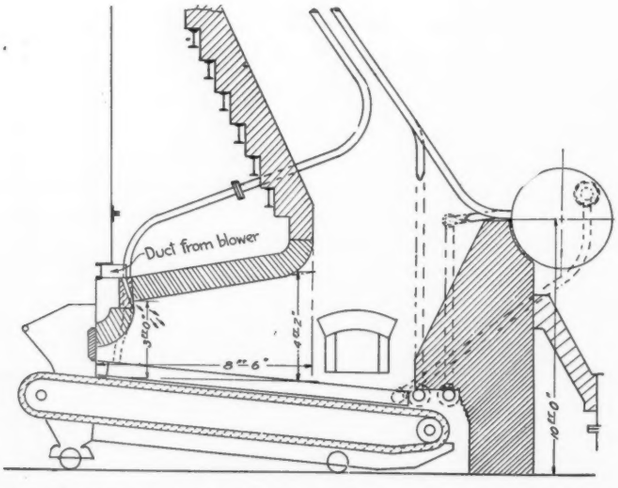


Fig. 1—Natural-draft, chain-grate stoker with side-wall boxes and water-back.

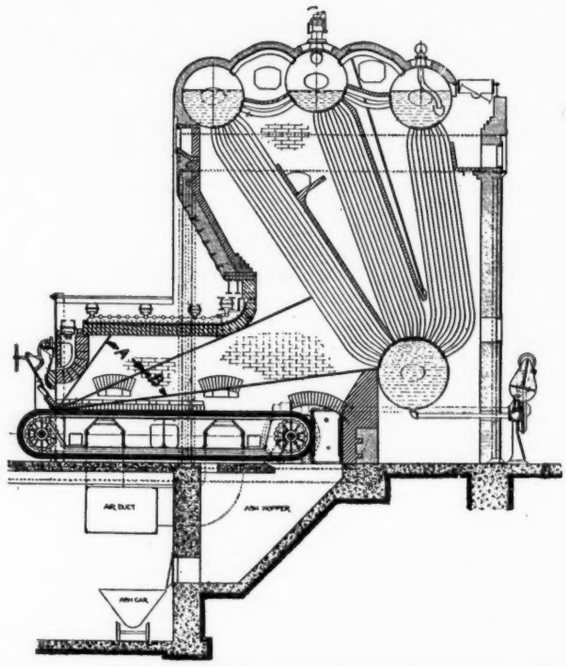


Fig. 2—Graphical method for determining correct arch length for anthracite-type furnace.

The modern, forced-draft, chain-grate stoker is most versatile as it will easily handle a wider range of fuels than perhaps any other type of stoker. The following fuels are commonly handled:

Anthracite	Bituminous (free burning)
Coke Breeze	Sub-Bituminous
Semi-Anthracite	Lignite

With stokers of this type, the design of furnaces and the extent of water-cooling it is possible to use are influenced by the fuel to be burned. The volatile matter and fixed carbon content are the principal governing factors. Because these differ widely in the fuels listed, it is necessary to divide the discussion into two sections; one dealing with anthracite and coke breeze and the other with bituminous coals and lignite.

The burning of anthracite and coke breeze is practically limited to the forced-draft, chain-grate stoker. It was for the burning of the smaller sizes of anthracite that this type of stoker was originally developed by Eckley B. Coxe in 1892-1893. Coke breeze was first burned successfully on this type of machine in 1916.

Steam plant sizes of anthracite range downward in size from pea coal to No. 4 buckwheat and culm. The larger sizes are usually burned on hand-fired grates. No. 3 and No. 4 buckwheat and culm are burned on the forced-draft, chain-grate stoker. The following is a typical proximate analysis of the smaller sizes of anthracite coal:

Moisture	8.00 per cent
Volatile Matter	8.07 per cent
Fixed Carbon	74.30 per cent
Ash	9.63 per cent
B.t.u. per lb. as fired	12,850

In the manufacture of coke considerable quanti-

ties of fines result. By screening, these fines, known as breeze, are separated from the larger and more valuable coke. The following are typical proximate analyses, of this breeze, as taken from actual test data:

	Gas-House Breeze	*By-Product Breeze*
Moisture	5.20 per cent	7.00 per cent
Volatile Matter	7.32 per cent	4.71 per cent
Fixed Carbon .	73.90 per cent	72.64 per cent
Ash	13.58 per cent	15.65 per cent
B.t.u. per lb. as fired ...	10,750	11,000

The above analyses, of both anthracite and coke breeze, indicate that the volatile matter content of these fuels is extremely low. It is because of this characteristic that ignition difficulties are inherent unless proper furnace design is provided.

To obtain positive ignition it is necessary to employ refractory arches to provide the required radiant heat for kindling the incoming green fuel and also to shield it from the cooling effect of the boiler surfaces. The arrangement and design of the arches will fall into one of the three following classifications: front, combination or rear arch furnace design. The description of these three types of furnaces will be discussed separately as the design principles differ materially.

The early installations were all provided with the front arch design of furnace. Indifferent success, in a number of cases, made it evident that a definite relation between the length of arch and its height above the grate must be established, if constant and positive ignition were to be obtained. Since ignition is dependent on the amount of heat radiated to the incoming fuel bed, it is obvious that this reflected heat must exceed that taken away by the relatively cold, water-heating surface. Furthermore, engineering judgment, based on experience, dictated that the minimum height of arch above the grate should not be less than 3 ft. 6 in. or more than 6 ft. 0 in. Having established the arch location within these limits, it now becomes necessary to provide sufficient length to give the proper amount of radiant heat. For determining

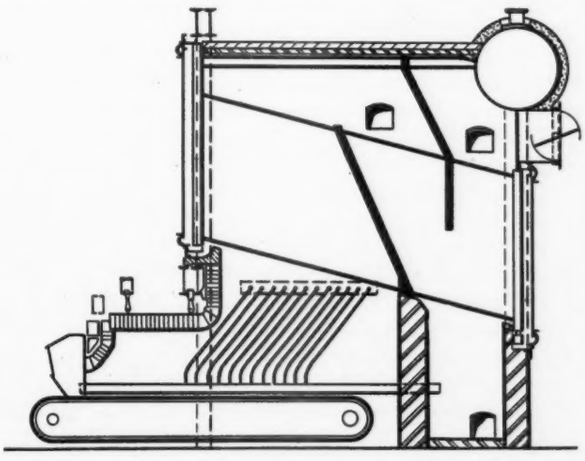


Fig. 3—Front-arch-type, anthracite furnace with side-wall cooling.

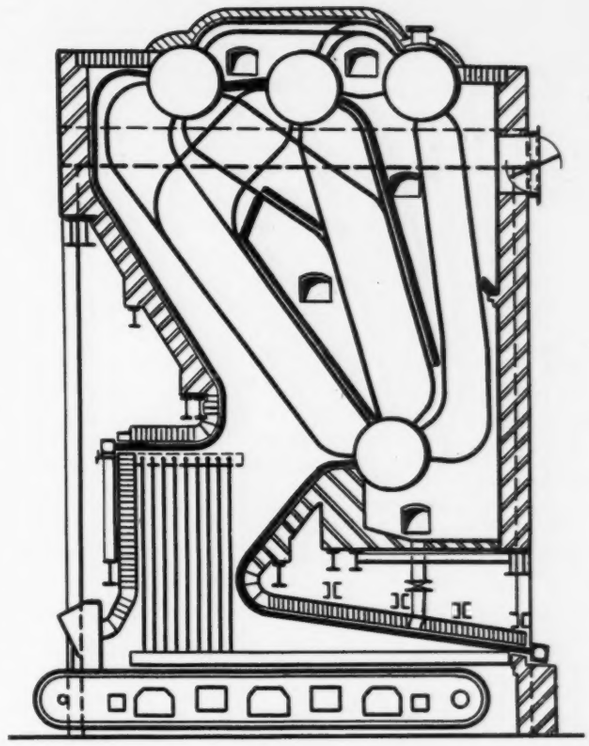


Fig. 4—Combination-arch-type furnace with water-cooled lining for small sizes of anthracite.

the relation between length and height of arch, a graphical solution may be used with certainty. In Fig. 2 is shown a front arch installation under a bent-tube type of boiler. The heat available for ignition is indicated by the angle A. The absorbed heat is indicated by angle B. When angle A is greater than angle B positive ignition is assured. When these two angles are equal the fuel kindles slowly and has a tendency to pull away from the gate as ignition becomes uncertain. Unless the operator takes steps to remedy this condition, by establishing a balanced draft or even pressure in the furnace, the fuel bed will cool rapidly. When angle A is smaller than angle B it will be practically impossible to ignite the fuel even though a pressure is maintained in the furnace. The obvious remedy, in this case, is the relocation of the arch or an entirely new design of furnace.

Because the front arch type of furnace is usually provided with just sufficient arch to assure ignition, the question of the addition of water-cooled lining must be given careful consideration, if failure is to be averted. Since the main object in providing this type of surface is to reduce maintenance costs, only those areas requiring frequent repairs should be protected. Any additional surface would be superfluous and fail to pay a return on the investment cost. In furnaces of this type there are three principal points of refractory maintenance: first, along the grate line due to clinker formations, which are removed by slicing with resulting damage to the refractories; second, the furnace side walls, beginning at a point about 1 ft. from the end of the arch and extending up to-

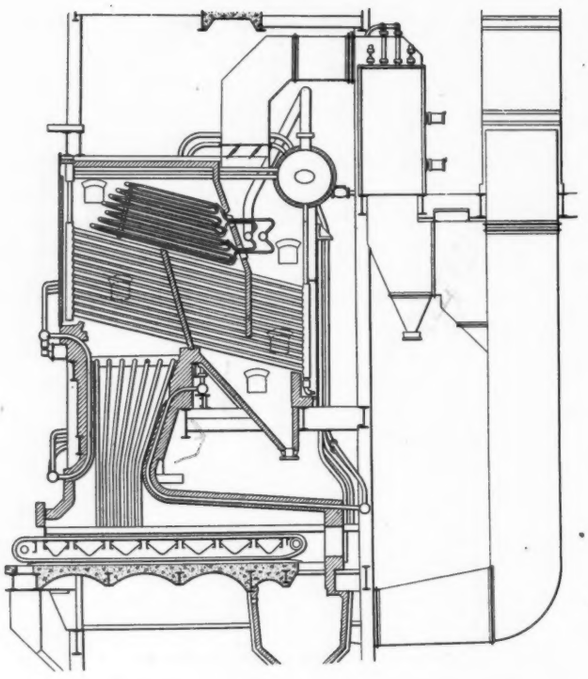


Fig. 5—Water-cooled, rear-arch-type furnace for small sizes of anthracite.

ward the boiler surface (this is due to flame and ash erosion); third, the nose of the arch. The application of side wall boxes, tied into the boiler circulation, will serve to eliminate the difficulties resulting from the too diligent use of a slice bar. A number of bare tubes, on relatively wide centers with refractory between them, will give adequate protection to the side walls and at the same time serve to reduce the maintenance on the nose of the arch. Of course if desired a refractory-covered, water-cooled arch may be installed but its cost will be quite high, and it is doubtful if it can be economically justified. In Fig. 3 is shown an installation provided with side-wall cooling of the type described above. In this case the side-wall boxes serve also as the lower side-wall header and thereby reduce the cost of the installation. The cooling surface is composed of 4 in. bare tubes spaced on 7 in. centers and the intermediate refractory is carried out to the center line of the tubes. The tubes are sloped so that they cover that portion of the wall under the arch which is subjected to severe erosion. By installing the wall protecting surface in this manner, ignition is not affected, as practically none of this surface "sees" the incoming fuel. Furthermore very little heat is taken away from the arch and thus the amount available for ignition purposes is not materially reduced.

When burning No. 4 buckwheat and "river coal" in the front arch type of furnace, the ash pit losses were exceedingly high. This was due to the fact that when sufficient air pressures were carried to establish the best fuel bed conditions, a large percentage of the fines were blown from the front of the grate and fell into the ash pit. To overcome this difficulty, the combination arch type of furnace was designed. The first one had only a stub or short protecting arch which prevented the fines, that were blown from the front end of the fuel bed, from falling into the ash pit. Because this arch was so short there was no reduction in the ash pit loss. The reason for this is that the fines were deposited on the ash covering the rear end of the stoker and were then discharged into the pit, before being ignited and burned. However this short arch did serve a useful purpose in that it aided in mixing the leakage air, from the ash pit, with the richer gases from the front end of the stoker. The next design step was to increase the length of the rear arch so as to cover a greater portion of the stoker surface. With this arrangement, the fines blown from the front end of the grate fall on a portion of the fuel bed which is still "alive." The result is that these particles are ignited and are burned under the arch at the rear end of the furnace. The percentage of combustible, in the ash pit refuse, is accordingly lowered and the efficiency increased. The operation of this type of furnace has been extremely successful in burning No. 4 buckwheat and refuse fuel from culm banks which carry an extremely high percentage of dust.

In the latest development of the combination arch furnace the front arch is relatively short and is set about 12 ft. above the grate. The rear arch is carried forward into the furnace so that it covers approximately 50 per cent of the stoker length. The height of this rear arch above the grate is such that the velocity of gases flowing forward into the furnace is relatively high. The throat opening between the end of the rear arch and the front furnace wall is widened, so that the gas velocity, through this section, is reduced to approximately 65 per cent of that existing at the discharge opening of the rear arch. The result of this construction is considerable turbulence, because of the mixing of the gas streams, and a uniformly high CO_2 content in the gases leaving the furnace. In furnaces of this type, ignition is not secured by radiation from the front arch but principally by radiation from the nose of the rear arch, and because of the large percentage of ignited fuel that is blown from the rear end of the grate and deposited on the incoming green fuel, as it enters the furnace. The result is extreme stability of ignition over a very wide range of combustion rates.

Because of the stability of ignition in the combination arch type of furnace, it is possible to apply a larger percentage of water-cooled lining without impairing ignition. Stoker operation, with this construction, becomes stable and does not require constant attention to prevent the fires from going out. Fig. 4 illustrates an application of what may be considered as the maximum amount of cooling to be applied to a furnace for burning No. 4 buckwheat. In this instance, the bare tube type of construction is used for the front and rear arch of the furnace, as well as for the side and the upper front walls. The tubes forming the arches are on 10 in. centers and have refractory extending downward between them. In effect we then have

alternate ribbons of bare tube and refractory, forming the two arches. A water-cooled side wall box is used and this becomes the lower header of the tubes used for the side wall cooling. These tubes are bare with refractory material placed between them. Furnaces of this type have an extremely high availability factor because of the freedom from refractory maintenance.

The rear arch type of furnace is one in which the front arch is entirely omitted. Because of the design characteristics, it is necessary to modify the stoker operation particularly as to the method of supplying the air to the fuel bed. In the front and combination arch types of furnace, the bulk of the air is supplied at the front end, as the fuel enters the furnace, and the amount of air gradually reduced front to rear of the stoker. In the rear arch type of furnace the procedure is to increase the air supply gradually, as the fuel progresses into the furnace, until the greatest quantity is supplied through the next to the last stoker compartment. The arch is located so that the velocity of the gases issuing from it is high which causes a constant shower of burning fuel to travel toward the front end of the stoker. This, together with the radiation from the nose of the arch, assures positive ignition and further assists in reducing the combustible content of the refuse in the ash pit.

Here again we have a furnace design which lends itself to the application of the maximum amount of cooling. In Fig. 5 is illustrated an installation which has been in operation for more than two years. In this case a refractory-covered, water-cooled rear arch is provided. This refractory covering is carried to a point just above the arch throat. The side walls are also provided with bare tube surface which is partially refractory covered. The front wall of this furnace is also water-cooled, but in this instance no refractory covering is provided. There has never been an instance of ignition failure even though the unit is operated as low as 20,000 lb. of steam per hr, and as high as 100,000 lb. per hr. Furthermore, it is possible to so control the air supply that the CO_2 may be maintained at practically any desired value from 16 per cent downward. This is a particularly desirable feature as it provides a certain amount of steam temperature control, over a range of rating, that would not be possible otherwise. This plant is burning a grade of No. 4 buckwheat, regularly obtainable in the New York market, at a combustion rate of 35 lb. per sq. ft. of grate surface. The carbon in refuse seldom exceeds 20 per cent.

In the foregoing discussion of furnace designs, little or nothing has been said about coke breeze. The principles set down for burning anthracite apply equally to this fuel. Certain slight modifications are desirable because furnace temperatures are frequently higher when breeze is burned. Furthermore breeze is of an abrasive nature and provision should be made to minimize the scouring effect, on the refractories, of the dust particles carried in the furnace gases.

In the front-arch type of furnace the arch is rarely set lower than 4 ft. and where possible, it should be placed 6 ft. above the grate surface. The effect of this design feature is to provide more active furnace volume directly above the fuel bed. Because of the lowered velocity the cutting and abrasive action of the furnace gases is reduced. In some furnaces, because of this increased height, it will be necessary to provide greater arch length. Even though these design modifications are responsible for increased refractory areas, the maintenance factor is reduced and the availability is comparable to anthracite furnaces of the same type.

As with anthracite the amount of arch provided in this type of furnace is just enough to provide stable ignition. As a result the limitation on the amount of water-cooling surface which may be added is the same as with anthracite. Fig. 3 illustrates the extent to which water-cooling may be carried safely in furnaces of this type.

The combination arch type of furnace most generally used with coke breeze is shown in Fig. 6. In this design the rear arch extends forward sufficiently to cover the last two stoker compartments. The front arch is set from 5 to 6 ft. above the grate.

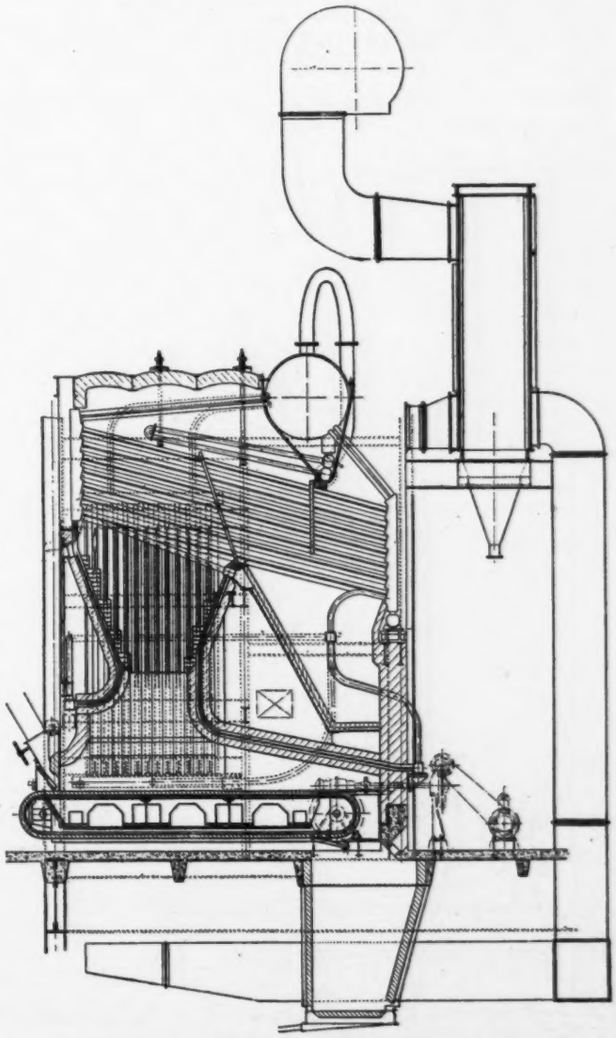


Fig. 6—Water-cooled, combination-arch-type furnace for coke breeze.

and is somewhat shorter than in the furnace just described. The stoker is operated with decreasing air pressures to the rear. The fines from the fuel bed are blown towards the ash pit and fall onto the grate in front of the rear arch. Because two compartments are provided under this arch, it is possible to burn out a greater percentage of the carbon than with the front arch type of furnace. Gas velocities in the furnace are relatively low and maintenance due to erosion is minimized.

The extent of wall cooling usually provided consists of side wall boxes along the grate line, and bare tubes covering the wall area between the ends of the two arches. In some cases, it is desirable to provide a water-cooled rear arch. This additional cooling may be readily justified in many installations as it does materially reduce arch maintenance. Furthermore it has the effect of increasing the availability factor and this is particularly important where there is limited installed boiler capacity, or where the operation is such as to require the unit to be on the line for long periods of time.

The installation shown in Fig. 6 was the first all water-cooled furnace to burn coke breeze successfully. The side walls and arches are completely covered with heat absorbing surfaces. The lower surface of the rear arch and side walls have a thin covering of plastic refractory. This unit has now been in service for a number of years and has performed remarkably well. When first started up, a greater portion of the refractory covering on the arch fell off. This has not been replaced because ignition was not impaired. With arches composed of alternate strips of tubes and refractory, we have a condition that is far more favorable, for ignition purposes, than the one just described. Therefore there should be no hesitancy in using this design where it is economically justified.

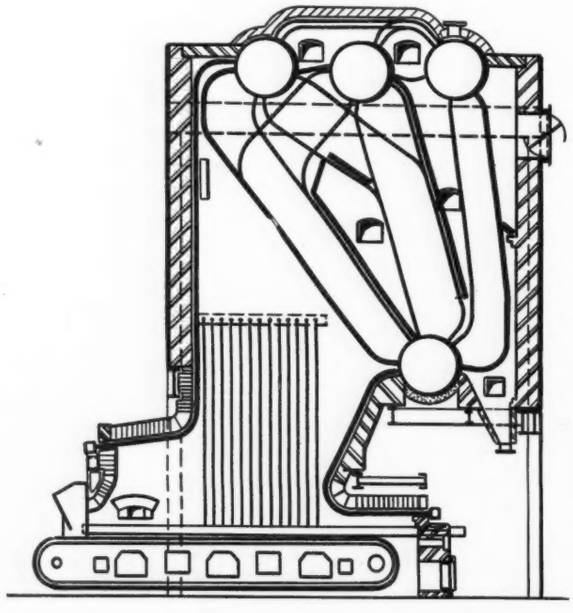


Fig. 7—Water-cooled, front-arch-type furnace for bituminous coal. Conventional water back replaced by water-cooled bridge wall.

When designing a rear arch furnace for coke breeze, the throat opening is made from 6 to 12 in. greater than with anthracite. The arch height, above the grate, is increased a like amount. As with anthracite, a completely water-cooled furnace may be provided. The most effective manner in which this surface may be installed is in the form of bare tubes on relatively wide centers, with refractory material between them.

High volatile coals, having "free-burning" characteristics, are most suitable for the forced-draft, chain-grate stoker. Into this classification fall the mid-western bituminous and sub-bituminous coals and also the lignites. Generally these have a relatively high sulphur content, they contain varying percentages of pyrites, and as a result the fusion temperature of the ash ranges between 1,800 and 2,300 deg. fahr.

Due to the high volatile content as well as the long flaming characteristics of these coals, it is necessary to provide greater furnace volume than required for either anthracite or coke breeze. Furthermore, because of this increased volatile content, kindling of the green fuel is more easily accomplished. As a result it is possible to increase the height and reduce the length of the arch. However, the increase in furnace volume in earlier installations introduced new problems in operation. Stratification of the furnace gases and smoke were encountered. This stratification continued through the boiler passes and resulted in reduced combustion efficiency. Examination of the furnace gases disclosed a decided lack of oxygen at the front end of the furnace, where the volatile matter is quickly driven out of the coal. At the rear end, where the fixed carbon is burned, there was an excess of oxygen. To secure a uniformity of gases leaving the furnace, and at the same time to insure completeness of combustion with a minimum amount of excess air, it was necessary to provide turbulence. This led to the application of over-fire air, introduced at high velocity, in the zone where oxygen is deficient. This resulted in uniformly high CO_2, elimination of smoke and gains in efficiency. Because of the turbulence above the fuel bed, and the possibility of operating at relatively high and uniform CO_2, a high furnace temperature is obtained. This, together with the fluxing and slagging action of the ash in the coals sets up conditions which cause severe erosion of the refractory furnace linings. In order to minimize this difficulty it is necessary to consider the application of water-cooled linings in all installations of this type.

As with anthracite, the bituminous installations are divided into front, combination and rear arch design. Regardless of the type of arch construction, over-fire air is considered an essential feature and should always be used.

All of the earlier installations were of the front arch construction. They were usually provided with water backs and sometimes had side-wall boxes included. Refractory maintenance was high even though the operation was carried out with

relatively large percentages of excess air, to reduce the furnace temperatures. There was hesitancy on the part of operators to accept furnace cooling, as they feared a return of smoke because of the chilling effect on the furnace gases. In one or two installations a few bare tubes were installed in the furnace side walls. Maintenance was immediately reduced and no operating difficulties were encountered. As a result, complete cooling of the side wall was soon tried out and arch cooling then followed. The resulting reduction in maintenance easily justified this more costly design, so that now there are many plants operating with the maximum amount of cooling that it is possible to install. In Fig. 7 is shown a front arch type of furnace provided with complete water cooling. The arch is composed of bare tubes spaced on relatively wide centers, with refractory material projecting between them to the centerline. The side wall tubes are vertical and spaced much the same as those in the arch. The conventional water-back is omitted, and in its place is a stub arch which is water-cooled. In some installations, where it is desired to reduce costs, this arch is omitted and the water back substituted.

The combination arch type of furnace has been used in a number of installations, with considerable success. In this design the amount of refractory surfaces exposed to the furnace gases is quite large, and unless suitably cooled will be short lived. This condition persists even though these arches are placed quite high above the fuel bed. The rear arch is from 2 to 3 ft. and the front arch from 1 to 2 ft. higher than the maximums established for anthracite design. The installation shown in Fig. 8 has now been in operation for a number of years, and the results obtained are excellent. Outage, because of wall failure, is practically unknown. In this design the rear arch practically covers three stoker compartments. The front arch extends over the same number. These two arches are composed of bare tubes on relatively wide centers with refractory between. In addition the upper section of both the front and rear furnace walls is water-cooled. Side wall water-boxes are provided at the grate line. These boxes are the supply headers for the side wall cooling which extends some distance above the furnace throat. Illinois, Iowa, Kansas, Missouri and Arkansas coals have all been burned in this furnace, with results that exceed those obtainable on other types of furnaces, in the same plant. This gain is due, in part, to the water-cooling and in part to the arch design.

As yet no combination arch installations, for bituminous coals, have been made along the lines shown in Fig. 4. However, there is no reason why they should not be successful even though the features incorporated are principally designed to care for fine coal, which is easily blown off the grate and carried into the boiler setting.

The use of the rear-arch furnace for burning bituminous coals is increasing. At first, because of the large refractory arch projecting into the furnace, and the apparently small throat through

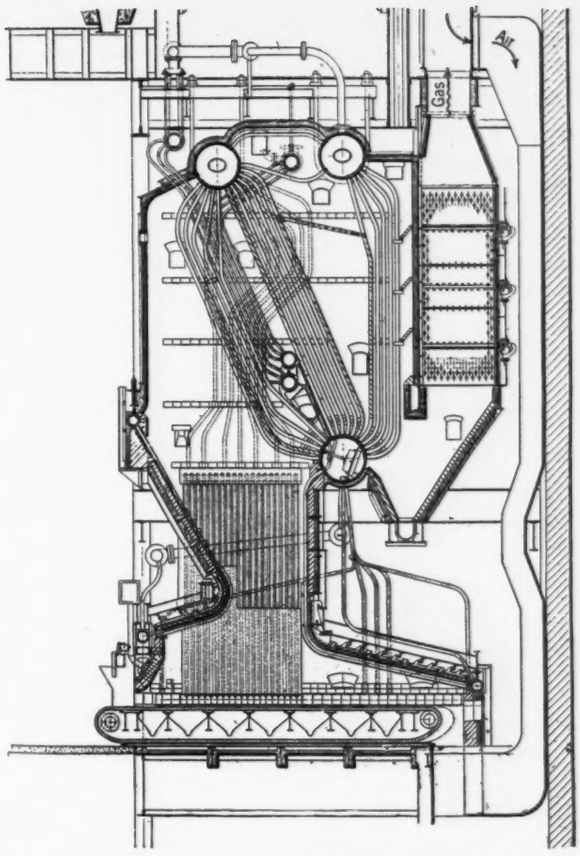


Fig. 8—Completely water-cooled, combination-arch-type furnace for mid-western coals.

which the gases pass, it was thought that maintenance would be prohibitive. However, it has been demonstrated in several installations that the cost of upkeep of this design is no higher than the others, and at the same time improved results are obtained. This construction lends itself to the application of water-cooling equally as well as the combination type of furnace. The construction, in this case however, is somewhat simpler and therefore less costly. In Fig. 9 is shown an actual installation which has been in operation for more than eighteen months, without any expenditures for furnace maintenance. This in spite of the fact that the maximum possible amount of furnace cooling was not applied. As shown in the illustration a water-cooled rear arch is provided. The front wall of the furnace is water-cooled except for the short refractory closure arch, between the fuel gate and the lower end of the front wall. The riser tube from the side-wall water box is brought back into the furnace, and serves to partially cool the side wall. More effective side wall cooling may be had by installing vertical tubes covering the width of the furnace throat. This latter addition of surface would be desirable if higher ratings than those required now, become necessary. From the appearance of this furnace it would seem that volume was being restricted even though practice indicates that a larger one is required. However, the fact that the rear arch furnace has turbulence

built into it, and in addition, is provided with overfire air, makes it unnecessary to provide the same amount of volume as is required with the older front arch design.

The foregoing discussion indicates that better efficiencies are obtained with the rear arch furnace than with either the combination or front arch type, regardless of the fuel burned. One of the reasons for this is that a lower percentage of excess air is required to complete combustion, because of the turbulent mixing obtained, when the gases pass, at high velocity, through the relatively nar-

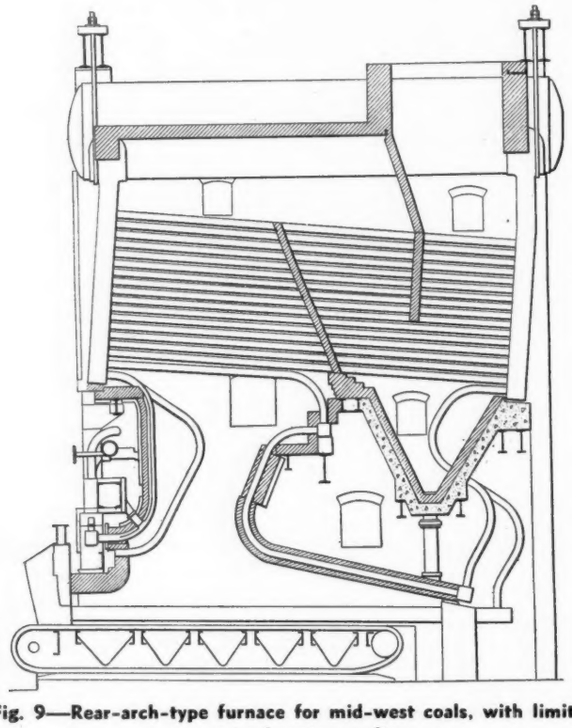


Fig. 9—Rear-arch-type furnace for mid-west coals, with limited amount of water cooling.

row furnace throat. Another reason is that the combustible in refuse is lowered because any fines blown up from the fuel bed are deposited at the front of the furnace, and are re-ignited and burned with the incoming fuel. Similar conditions exist in furnaces of the combination arch type, but not to the same degree. The amount of excess air required for combustion may be the same, but the carbon in refuse loss will be somewhat higher.

Because of high furnace temperatures, resulting from operation with low percentages of excess air, furnace wall cooling is desirable. The illustrations indicate the extent to which this may be applied in the various types of furnaces. Many complete refractory settings are being operated successfully, but the inherent limitations of refractory materials frequently make it impossible to take full advantage of the design provided, except at the sacrifice of maintenance and availability. The installation of suitable water-cooled surfaces removes these limitations and thus permits maximum flexibility with respect to load conditions while assuring minimum maintenance costs.

1932 Coal and Ash Handling Report

Issued by Prime Movers Committee

The Prime Movers Committee of the N. E. L. A. has issued the above report for 1932. It includes a statement on the commercial by-product possibilities of pulverized coal ash in which data are given of laboratory and field tests which show both successful and unsuccessful results, the latter predominating. One by-product appears to have commercial value. Other subjects covered by the report are the location and arrangement of coal storage and handling equipment, including a description of a somewhat radical departure from usual practice, and central station methods of weighing coal. The latter statement analyzes the various methods and emphasizes the necessity of cleanliness, regular and thorough inspection, and frequent testing as essentials to accurate results.

Copies of this report, which is designated as N. E. L. A. publication No. 210, April 1932, may be obtained from the National Electric Light Association, 420 Lexington Avenue, New York. Price 25c to members; 40c to non-members.

American Society of Mechanical Engineers, 29 West 39th Street, New York, announce that addenda to the following sections of the A. S. M. E. Boiler Construction Code, will be available about August 1st, and can be obtained direct from the Society at the prices named:

Power Boiler Code—30c a set
Locomotive Boiler Code—10c.
Miniature Boiler Code—10c.
Low-Pressure Heating Boiler Code—15c a set
Unfired Pressure Vessel Code—30c a set
Rules for Care of Power Boilers—15c a set.

It is suggested that orders with remittance be sent them before July 15th.

L. Mundet & Son, Inc., Houston, Texas, manufacturers of Mundet "Jointite" Cork board, cork pipe covering and cork insulation, announce that they have established a sub-branch office at 4600 Gaston Ave., Dallas, Texas. Mr. A. A. Stone, formerly one of the sales engineers of the Houston office, has been transferred to Dallas, where he is now in charge.

The Mason Regulator Company, Boston, Mass., manufacturers of automatic regulating and control equipment, announces the opening of a new office in Chicago at 1525 East 53rd Street. The office will be in charge of William P. Gearon who has represented the Mason Company in Chicago for the past five years.

This office will also be occupied by the Neilan Co., Ltd., (a Division of the Mason Regulator Co.). The Neilan representative is Arthur J. Foley.

The Performance Characteristics of Centrifugal Boiler Feed Pumps

By

J. G. MINGLE, Indianapolis, Indiana

This is a very practical discussion of centrifugal boiler feed pump performance. The author gives equations and charts for the determination of pump capacity, total dynamic head and the several heads of which it is comprised, i.e., pressure head, friction head, velocity head and static head. Equations are also given for ascertaining the various pump efficiencies, i.e., mechanical efficiency, hydraulic efficiency, volumetric efficiency and total efficiency. This discussion will be continued in Part II.

THE performance characteristics of a centrifugal boiler feed pump constitute a set of three curves which express graphically: (1) the relation between the head created and the capacity delivered, generally called the head-gallon, head-delivery, or head-capacity, curve; (2) the brake horsepower involved and (3) the efficiency. These curves are the plotted results of a test made on the pumping unit on the testing floor of the factory in which it has been manufactured. The operating conditions for which the pump was designed, and to which it will be subjected when it is finally piped up to the boilers which it is to serve, are duplicated as nearly as possible when the unit is tested. The test then is a trial run of the pump in which the relation between the head created in terms of feet of water, or pounds per square inch gage pressure, and the capacity delivered in terms of gallons per minute is established, the amount of power involved in driving the pump under various conditions of head and the corresponding capacity is observed, and the efficiency of the operating pump computed from the head, capacity and brake horsepower data gathered from the various points. In other words, the test together with the characteristics give an accurate series of results from a series of arbitrarily imposed operating conditions from which the performance of the pump under actual operating conditions can be judged.

The required capacity of a centrifugal boiler feed pump is expressed in terms of gallons of water discharged, or delivered, per minute (g.p.m.) and is determined from the horsepower of the boiler, or boilers, which it is to serve. Since a boiler horsepower (boiler hp.) is equivalent to the evaporation of 34.5 lb. of water per hr. from a temperature of 212 deg. fahr. to steam at atmospheric pressure and also since one U. S. gallon of water weighs 8.34 lb. at 62 deg. fahr., then:

$$\text{g.p.m.} = \frac{\text{boiler hp.} \times 34.5}{60 \times 8.34} = 0.0069 \text{ boiler hp.} \quad (1)$$

In practice, however, the equivalent of evaporation factor is generally assumed at 30 lb. of water per hr. Whence:

$$\text{g.p.m.} = \frac{\text{boiler hp.} \times 30}{60 \times 8.34} = 0.006 \text{ boiler hp.} \quad (2)$$

In general: 1000 lb. of water evaporated per hour = 2 g.p.m.

The capacity should be based on the maximum, or ultimate, horsepower of the boiler, or boilers, which the pump is to serve. For example, for a single boiler with a normal rating of 1000 hp. and a maximum rating of 300 per cent, the pump should be selected for a capacity of $1000 \times 3.00 = 3000$ boiler hp. $\times$ 0.006 = 180 g.p.m. Again, for two boilers, either in a single or battery setting, each with a normal rating of 1000 hp. and a maximum rating of 250 per cent, the pump should be selected for a capacity of $1000 \times 2 \times 2.50 = 5000$ boiler hp. $\times$ 0.006 = 300 g.p.m.

In determining the capacity of a centrifugal boiler feed pump, it is better by far to err on the side of safety, within reason, rather than to have an insufficient capacity. When the capacity is excessive, the pump can be throttled at the discharge and the amount of water delivered thereby regulated. However, when the capacity of the pump is insufficient, it becomes necessary to either increase the speed of the unit, which is not always possible, or provide another pump to operate with it in parallel in order to provide the required amount of water. When the pumping unit is operated at a constant speed, as with a constant speed motor, a second pump is always required to provide more water. With steam turbine driven and variable-speed, motor-driven units, it is possible to increase the capacity of the pump within a certain definite limit by increasing the speed of the driving unit. With constant-speed, motor-driven units, however, the capacity of the pump is fixed at any point on the head-gallon curve but it will vary as the head varies.

Erring on the side of safety in determining the

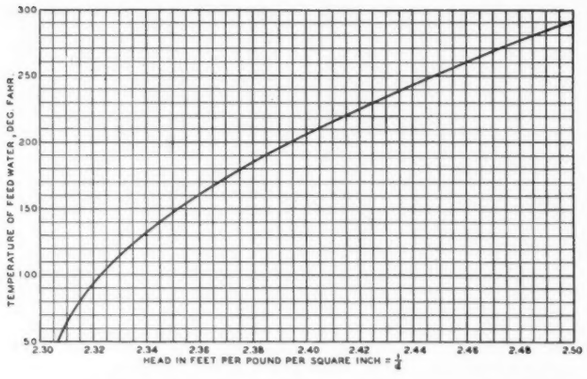


Fig. 1—Relation between feedwater temperature and head in feet per pound per square inch gage pressure.

capacity of a centrifugal boiler feed pump should be done within reason, otherwise it will become necessary to throttle the discharge to the extent that the pump will be compelled to operate at an excessively high head, especially on a steep head-gallon curve, and also at a correspondingly relatively low efficiency with, as a consequence, a relatively high brake horsepower.

From a capacity standpoint, it is oftentimes more economical to provide two pumping units: one unit being used when the boiler, or boilers, are being operated at their normal rating and the other being placed on the line and operated in parallel with the ordinarily used pump when the boiler, or boilers, are operated at higher rating. This system insures either one or both pumps operating both at a reasonable and proper head and also at the highest efficiency for which they were designed with a consequent relatively low brake horsepower.

The head of a centrifugal boiler feed pump may be expressed either in terms of pounds per square inch gage pressure, or feet of water at the corresponding temperature. When a pump is operating at a certain speed, it must develop, or create, a head equivalent to the total dynamic head of the outside system plus a differential head to force the water into the boilers. The head developed by the pump is equal to the head imparted to the water by the impellers less the losses of head within the

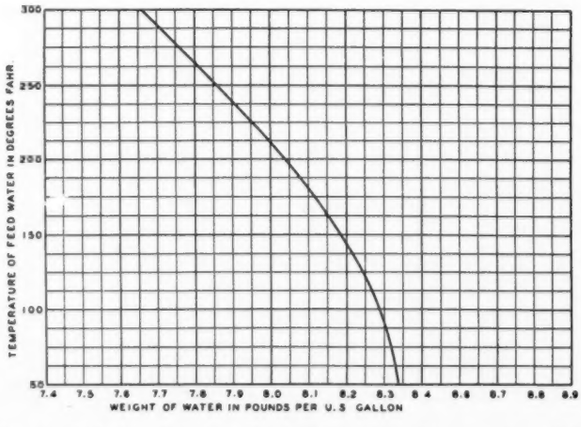


Fig. 2—Relation between feedwater temperature and weight per gallon.

pump due to friction, entrance losses, eddy losses, cavitation, etc.

The total dynamic head (H_d) of the outside system against which the pump is to operate, expressed in terms of feet of water at 62 deg. fahr. and sea level atmospheric pressure, is equal to the algebraic sum of the pressure head due to the boiler pressure (h_p) which is equal to the boiler pressure (P_b) expressed in pounds per square inch gage at the corresponding temperature (d), the suction lift (h_l) or suction head (h_s) as the case may be, the friction head (h_f), the velocity head (h_v) and the static head (h_e) and is given by the equation:

$$H_d = h_p \left(= \frac{P_w}{d} \right) + h_l - h_s + h_f + h_v + h_e \quad (3)$$

The expression for the pressure head due to the boiler pressure (h_p), or the equivalent gage pressure in feet of water, is developed as follows: At sea level atmospheric pressure, pure water weighs 62.4 lb. per cu. ft., or 62.4/144 = 0.434 lb. per sq. in., at 62 deg. fahr. (d). Then:

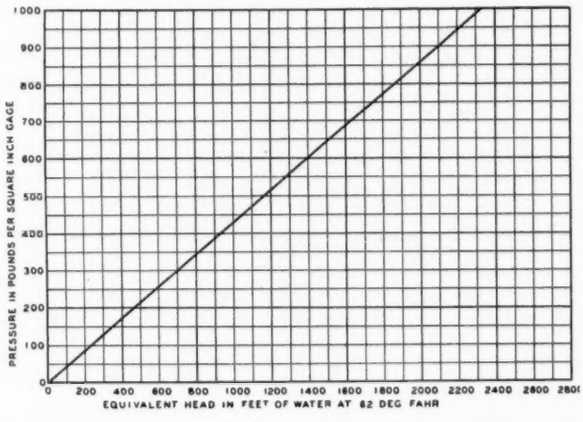


Fig. 3—Relation between gage pressure and equivalent water column.

$$P_w = 0.434\ h_p \quad (4)$$

and $$h_p = \frac{P_w}{0.434} = 2.31\ P_w \quad (5)$$

In general: $$P_w = d\ h_p \quad (6)$$

and $$h_p = \frac{P_w}{d} \quad (7)$$

The temperature of the feedwater may vary from a temperature of ordinary well water to a temperature as high as 280 deg. fahr. and, in special installations, even as high as 520 deg. fahr. Fig. 1 shows the relation between feedwater temperature and head in feet per pound per square inch gage pressure, or values of $\frac{1}{d}$ in equation 7. Fig. 2 shows the relation between feedwater temperature and weight per gallon and Fig. 3, the relation between

gage pressure and equivalent water column at 62 deg. fahr.

The pressure head (h_p) is usually determined from equation 7 by multiplying the boiler pressure by the $\frac{1}{d}$ factor corresponding to the proper feed-water temperature as given by equation 1. The value of the pressure to be substituted in equation 7 should be slightly in excess of the maximum boiler pressure, a differential of about 10 lb. usually being sufficient. Sometimes the total dynamic head is determined from equation 7 by allowing a differential pressure of 20 to 30 lb. above the maximum boiler pressure and properly correcting for density variation due to temperature. This practice is hazardous due to the fact that the sum of the friction, velocity, static, and other heads may be in excess of the differential pressure allowed. It is always the best practice to determine the actual values for the various heads and then make the necessary allowance for differential pressure as noted.

The friction head (h_f) may be determined from Figs. 4 and 5 which are based on the tables of friction of water in pipes as given in the Standards of the Hydraulic Society. These curves give the loss of head due to friction per 100 ft. of 15-year-old smooth iron pipe. To the actual length of the discharge pipe between the discharge of the pump and the entrance into the boiler should be added the pipe length equivalent due to valves, tees and elbows. Fig. 6 gives the equivalent length of straight pipe for valves, tees and elbows for various diameters of pipe. The curves shown in these three figures are based on ordinary iron pipe which has been in service for approximately 15 years. For new pipe the friction head of course is somewhat less and for old pipe considerably greater than the values shown. For approximate usage, the friction head for new pipe may be assumed at 0.75 and for old pipe 1.50 of the values shown by Figs. 4 and 5.

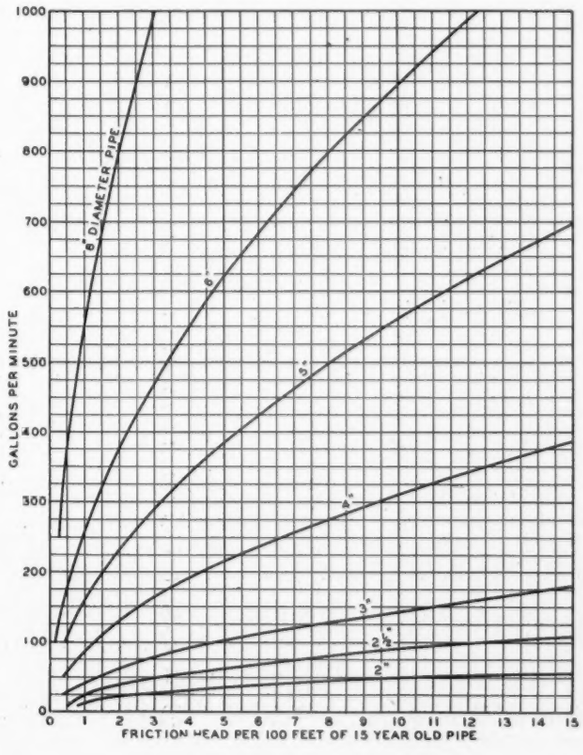


Fig. 4—Friction of water in pipes

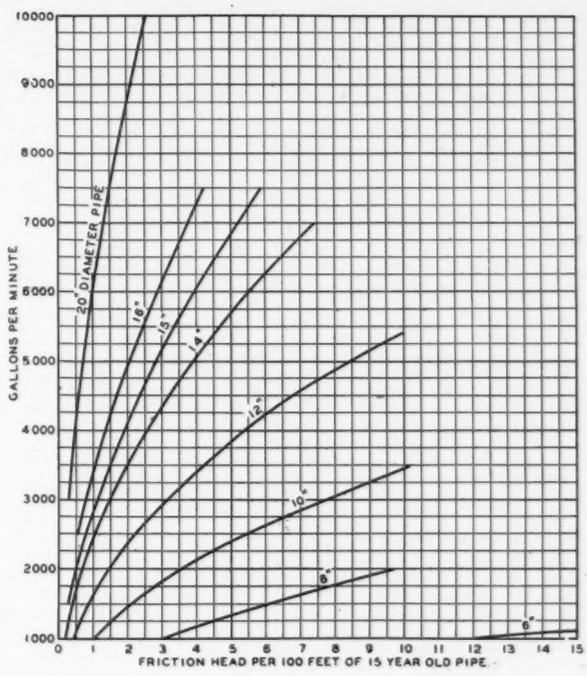


Fig. 5—Friction of water in pipes.

The velocity head (h_v) is the head necessary to accelerate the water, but its value is usually small and for all practical purposes may be neglected. However, it is a factor in the total dynamic head and should be considered in accurate testing. Fig. 7 gives the velocity head for various velocities of water. Fig. 8 gives the velocity of water in pipes of various diameters for various quantities of water.

The static head (h_s) is the difference in elevation between the center of the discharge opening of the pump to the center of the discharge pipe entering the boiler.

A suction lift condition for a centrifugal boiler feed pump is a rare condition except in cases where the temperature of the feedwater is low. When the suction lift condition does exist, it should be included in the total head and the distance should be taken from the lowest level of the free source of supply to the center of the suction opening of the pump. The suction lift is measured by means of a mercury gage. Fig. 9 shows the relation between inches of mercury gage column and the corresponding suction lift in feet of water.

The temperature of the feedwater entering the boilers varies between wide limits and may be as high as 520 deg. fahr. in special installations, but for ordinary practice seldom exceeds 280 deg. fahr. When the temperature of the water in the heater

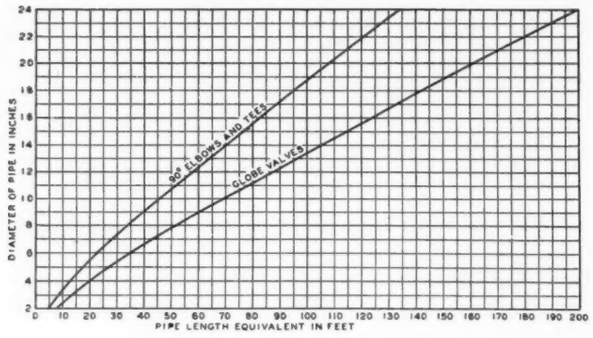


Fig. 6—Pipe length equivalent for elbows, tees and valves for determining friction head.

reaches 212 deg. fahr., a proper head must be maintained on the suction end of the pump at this, and higher, temperatures in order to prevent the feedwater from being transformed into steam and also to prevent the pump from becoming steam bound. In other words, a pressure must be maintained on the suction of the pump over and above that required to prevent the water from being transformed into steam at that particular temperature. Fig. 10 gives the required pressures on the suction of the pump for various feedwater temperatures.

The mechanical efficiency (E_m) of a centrifugal boiler feed pump is the ratio of the power actually delivered to the liquid (not by the liquid) to the power supplied to the pump. The power delivered to the liquid is equal to the product of the amount of water delivered (g.p.m.) plus the leakage (L), the unit weight of the feedwater at the indicated

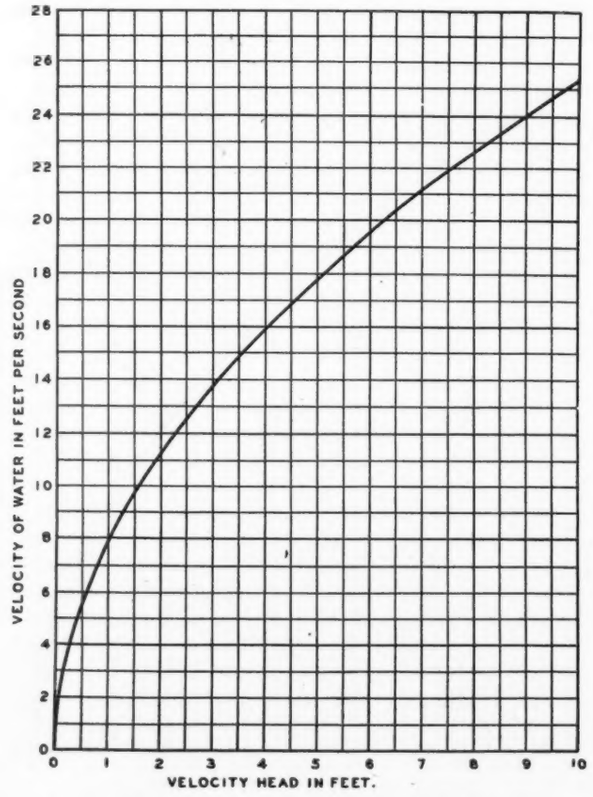


Fig. 7—Relation between velocity and velocity head.

temperature (w) and the theoretical head delivered by the impellers (h_t) divided by 33,000. The power supplied to the pump (b.hp.) is equal to the power supplied by the driving unit. Whence:

$$E_m = \frac{(\text{g.p.m.} + \text{L})\ \text{w}\ h_t}{33{,}000} \Big/ \text{b.hp.} \qquad (8)$$

The mechanical efficiency expresses the losses within the pump due to friction of the rotating element, etc. and is determined by the actual test on the testing floor.

The hydraulic efficiency (E_h) of a centrifugal

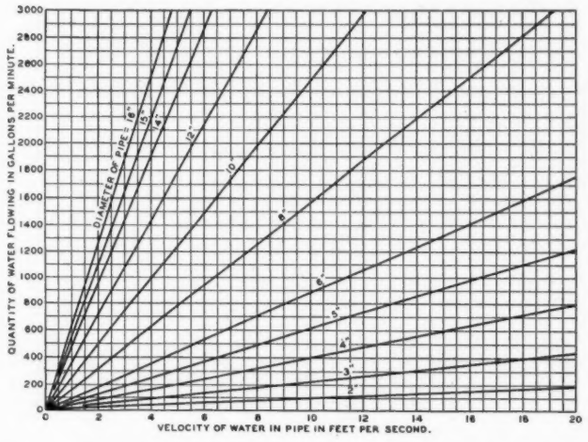


Fig. 8—Relation between velocity and quantity of water for various diameters of pipe.

boiler feed pump is the ratio of the useful power delivered to the water at the point of discharge where measured (h_a) to the power delivered to the water by the impellers at the point of exit from the impellers (h_t). The power delivered to the water is equal to the water horsepower (w. hp.) of the pump and the power imparted to the water by the impellers is equal to the product of the actual discharge by the pump (g.p.m.), the unit weight of the feedwater at the indicated temperature (w) and the theoretical head developed by the impeller (h_t) divided by 33,000. Whence:

$$E_h = h_a/h_t = \text{w.hp.} \Big/ \frac{\text{g.p.m. w}\ h_t}{33{,}000} \qquad (9)$$

The hydraulic efficiency is, in effect, the ratio of the total actual head which the pump develops to the theoretical head developed by the pump (the total actual head plus all losses).

The volumetric efficiency (E_v) of a centrifugal boiler feed pump is the ratio of the water actually delivered by the pump (g.p.m.) to that discharged by the impellers (g.p.m. + L). Whence:

$$E_v = \frac{\text{g.p.m.}}{\text{g.p.m.} + \text{L}} \qquad (10)$$

The total, or overall, efficiency (E_t) of a centrifugal boiler feed pump is equal to the product of the mechanical efficiency, (E_m), the hydraulic effi-

ciency (E_h) and the volumetric efficiency (E_v). Whence:

$$E_t = E_m \times E_h \times E_v \quad (11)$$

The total efficiency of a centrifugal boiler feed pump as generally used and without qualification is usually termed simply the efficiency of the pump and is equal to the ratio of the power output to the power input, or the water horsepower (w.hp.) to the brake horsepower (b.hp.). Whence:

$$E_t = \frac{\text{w.hp.}}{\text{b.hp.}} \quad (12)$$

The water horsepower (w.hp.) is equal to the product of the gallons-per-minute displacement (g.p.m.), the unit weight of the feedwater at the indicated temperature (w) and the total dynamic head in feet against which the pump operates (H_d) divided by 33,000. Whence:

$$\text{w.hp.} = \frac{\text{g.p.m. } H_d}{33{,}000} \quad (13)$$

For clear water at 50 deg. fahr., w = 8.34 lb. per gal. Hence, for this condition equation 13 reduces to:

$$\text{w.hp.} = \frac{\text{g.p.m. } H_d}{3960} \quad (14)$$

The value of w, however, varies inversely as the temperature. Fig. 2 gives the unit weight per gallon for various temperatures noted. In applying equation 13, care should be taken that either the proper values be supplied for w, or the dynamic head (H_d) when expressed in pounds per square inch be properly corrected as given in Fig. 1.

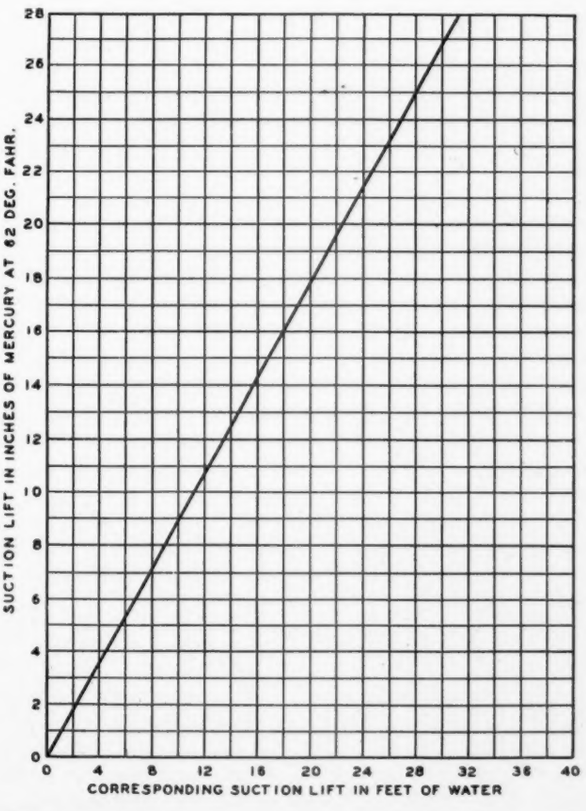


Fig. 9—Relation between inches of mercury and feet of water.

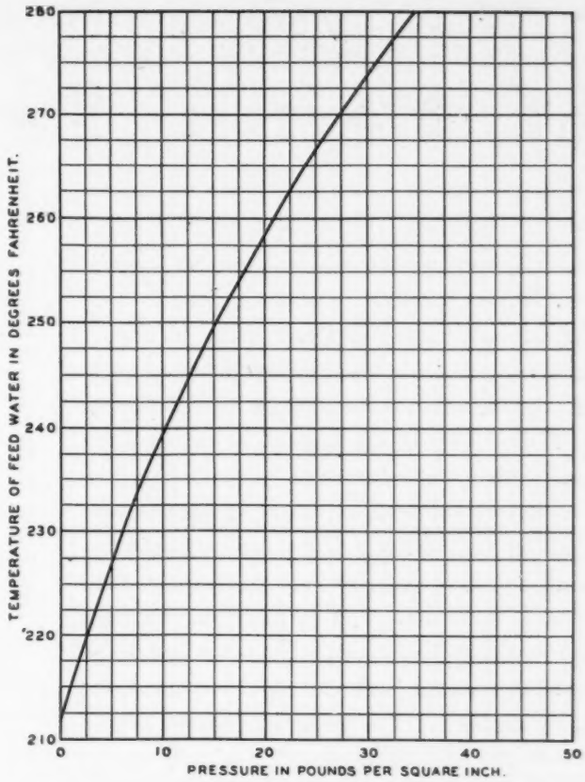


Fig. 10—Pressure necessary to maintain on suction end of centrifugal boiler feed pumps for various feedwater temperatures.

The brake horsepower of an alternating current motor (b.hp.$_{acm}$) is equal to the product of the volts (V), the amperes (A), the power factor of the motor (Cos θ), the motor efficiency (E_{acm}), and K (1 for single phase, 2 for two phase and 1.73 for three phase) times 1.34. Whence:

$$\text{b.hp.}_{acm} = 1.34 \times V \times A \times \text{Cos}\,\theta \times E_{acm} \times K \quad (15)$$

The brake horsepower of a direct current motor (b.hp.$_{dcm}$) is equal to the product of the volts (V), the amperes (A) and the motor efficiency (E_{dcm}) times 1.34. Whence:

$$\text{b.hp.}_{dcm} = 1.34 \times V \times A \times E_{dcm} \quad (16)$$

In general, the brake horsepower involved in centrifugal pumping (b.hp.) is equal to the water horsepower (w.hp.) divided by the efficiency of the pump (E_t). Whence:

$$\text{b.hp.} = \frac{\text{w.hp.}}{E_t} = \frac{\text{g.p.m. w } H_d}{33{,}000\ E} \quad (17)$$

The overall efficiency of a centrifugal boiler feed pumping unit (E_o) is equal to the efficiency of the pump (E_t) times the efficiency of the driving unit (E_d). Hence:

$$E_o = E_t \times E_d \quad (18)$$

The Economics of Electric Power Production

By JOHN T. FARMER
Consulting Engineer, Montreal, Canada

In this, the second and concluding part of Mr. Farmer's article, the discussion is continued with a consideration of the effects of variations in basic costs on the relative overall economies of steam and hydro power. The next factor considered is the relative cost with both systems of secondary power, i.e., power supplied in off-peak periods, the demand for which can be met without increasing plant capacity over that required for primary demands. The author then discusses the possibilities of combined steam and hydro systems viewing first one and then the other as an auxiliary. The latter part of the article is devoted to a discussion of the problems of water storage which is followed by a general summary of conclusions reached with respect to both systems and the various combinations possible.

THE figures and diagram presented in Part 1 of this article refer to a specific set of conditions which were suggested as typical. Recognizing that no two sets of conditions encountered are exactly similar, the next step is to examine the effect of variations in these basic conditions.

In considering a hydro-electric installation the principal factor which will affect the ultimate cost of power obtainable is the cost at which the installation can be carried out. In the typical example under consideration this cost was taken at $150 per kw. of capacity. This unit cost is a very variable quantity. Examples can be found where this cost works out at $75 per kw. or even lower. On the other hand, special features of topography, or special difficulties of construction may run this cost up to $200 or even $300. The corresponding figure for a steam plant does not show as large variation, the usual limits being of the order of 20 to 25 per cent above or below the typical value of $100. Such variations, however, are overshadowed by what is in this case the most important variant, the cost of coal delivered at the plant.

This figure is liable to vary from $2 or even less per ton in locations adjacent to coal fields, to $8 or $10 where high transportation charges are incurred.

The effect of these variations on ultimate costs of power production is shown in the diagram, Fig. 2, wherein the total production costs are plotted in relation to load factor as in diagram, Fig. 1, but for a series of costs of installation in the case of hydropower and for a series of delivered costs of coal in the case of steam power.

It will be noticed that for some extreme values of these factors, one or the other system is evidently more advantageous throughout the whole range of load factor.

For example, if coal can be obtained for $1.80 per ton, and a hydro plant cannot be installed for less than $200 per kw., a steam plant will afford cheaper power at any load factor. On the other hand, if it is estimated that a hydro installation can be made for $120 per kw., such a plant will provide cheaper power at any load factor irrespective of the price of fuel.

In general, however, there will be found to be a load factor at which the steam and hydro costs equalize. Below this point the steam plant will show lower power cost; above it the hydro system will make the better showing.

To take a concrete example, a steam plant using coal at $4 per ton, and a hydro plant costing $175 per kw. will show equal overall power costs at a load factor of 33.5 per cent. If the demand to be met is mainly of a domestic character, so that the load factor is only 25 per cent, the steam plant will show a saving in total production cost of $10,000 annually, as compared with a hydro system. If, on the other hand, the demand load includes a considerable proportion of industrial load so that the load factor is brought up to 50 per cent, the condition will be reversed and the hydro system will show a saving of $20,000.

In the case of an isolated supply where the load factor is not very high and does not appear likely to improve, it frequently proves more economical to generate the power by steam, rather than to undertake a hydro development even though the water power is available and the conditions are favorable for its development at reasonable cost.

In any power system, either steam or hydro,

when the distribution includes what are known as essential services, an important object is to render the service free from interruption as far as humanly possible. This is done by duplicating all items of equipment whose outage, either accidental or premeditated, would reduce the plant capacity below the possible demand. The aim is to make provision for carrying the maximum load with the largest single piece of apparatus temporarily out of service.

The precise proportion of this increase in equipment, of course, varies with the set-up of the plant. Taking the average of a large number of existing steam plant systems of this class, it is found that the average excess of capacity provided is about 40 per cent of the maximum demand recorded.

The provision of such excess capacity, of course, involves an increase in the initial investment; and this in turn affects the carrying charges and the total cost of the power production. The modification of the total cost resulting from the provision for continuity of operation can be readily determined by the application of the principles used in drawing up the diagram, Fig. 1.

Applying the analysis to the typical case represented in Fig. 1, if the installed capacity of the steam plant is increased 40 per cent, this will mean an increase of investment of $200,000. This will be subject to the same charges as before for interest, depreciation insurance, taxes, etc., which means an increased carrying charge of $28,000 annually. The irreducible annual charges will therefore be increased from $93,308 to $121,308; and the graph of these charges on a load factor base will be a horizontal line parallel to that for the plant of net 5,000 kw. capacity.

The provision of the additional equipment will not appreciably affect the maintenance, operating labor, or supplies consumed so that the graph including these items will be an inclined line parallel to that previously developed, but starting from the higher point at zero load factor.

This modified graph of total production cost is shown in diagram, Fig. 3. It is interesting to note that the provision for security of continuous output entails an increase of total or average cost varying from 30 per cent at zero load factor to 10 per cent at 100 per cent load factor; or fully 20 per cent at the load factors commonly encountered.

This is the serious element of power production cost, which it is sometimes sought to alleviate by the interconnection of distributing systems with a view to reducing the total spare generating capacity.

A similar analysis can be applied in the case of hydro power. In this instance the necessity for reserve capacity only applies to the machinery and not to the permanent works. It does, however, apply to the transmission line which is a vulnerable part of the system in view of wind storms, sleet and lightning. In the typical case under review the additional investment necessary to safeguard the service at the center of distribution could be set down somewhat as follows:—

Increased machine capacity 40 per cent of 5,780 kw. @ $40..	$92,480	
Duplication of transmission line	200,000	
		$292,480

The annual charges on this investment, on the same basis as before, becomes:—

Interest		
7 per cent of $292,480	$20,474	
Depreciation		
1 per cent of $92,480	925	
3 per cent of $200,000	6,000	
Insurance, taxes, etc.		
1 per cent of $292,480	2,925	
		$30,324
Allowance for patrolling and maintenance of the second transmission line	5,000	
		$35,324

Thus the irreducible fixed charges at any load factor amount to $150,043. The energy increment charge follows an inclined line parallel to the graph previously developed but starting at the higher value at zero load factor.

It is interesting to note that making provision for security of service, the steam plant shows a lower cost of production up to 20 per cent load factor, instead of 15 per cent where no reserve equipment was provided for. This is largely due to the charges on the second transmission line.

Generally speaking, the length of main transmission line necessary is an important controlling factor in determining the relative desirability of hydro power in any given situation.

Primary and Secondary Power

In the discussion up to this point, attention has been confined to the comparative costs of primary power, that is to say power produced by generating systems set up for the purpose of meeting a recognized demand. In the operation of plants already installed, both steam and hydro, there often occur opportunities of utilizing additional energy in the distributing system without increasing the plant capacity for the express purpose of furnishing such energy. The plant capacity already installed is made use of for these "off-peak" loads at such times as it is not otherwise required. Power so generated and utilized is known as secondary power.

This situation occurs both in steam power and hydro-electric power systems. For such secondary power production, as there is no additional installation of plant necessary, the fixed charges incident to the production of a corresponding amount of primary power can be eliminated and the cost becomes simply the cost of generating the additional amount of energy.

As a concrete illustration of this, the typical installations for which Fig. 1 was drawn up, the assumption was that either plant, designed for a capacity of 5,000 kw., was only operated at 50 per

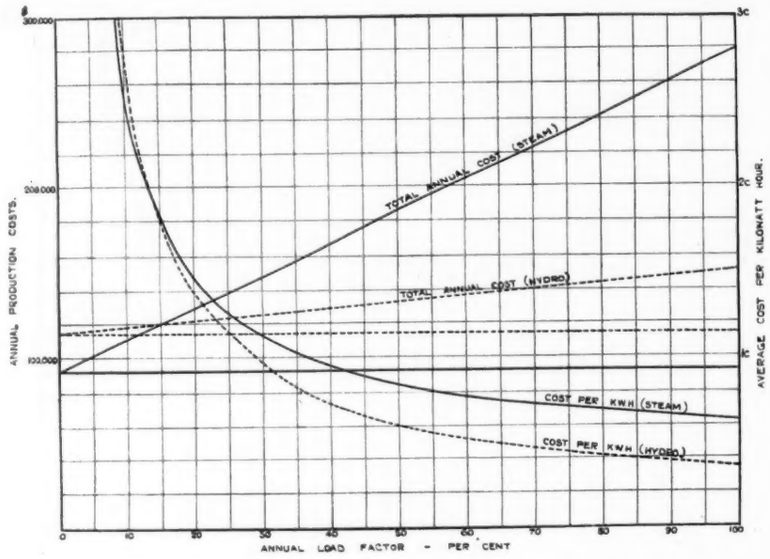

ANNUAL PRODUCTION COSTS.
AVERAGE COST PER KILOWATT HOUR.
TOTAL ANNUAL COST (STEAM)
TOTAL ANNUAL COST (HYDRO)
COST PER KWH (STEAM)
COST PER KWH (HYDRO)
ANNUAL LOAD FACTOR - PER CENT

Fig. 1—Annual production costs, steam and hydro 5000 kw. installation.

cent load factor, or at an average load of 2,500 kw. Suppose now that an outlet presents itself for the utilization of an additional average of 500 kw. or a total of 4,380,000 kw-hr. per year which, however, can be spread out so as not to encroach on any period when the load is otherwise in excess of 4,500 kw. The effect of such an additional load will be to increase the average load from 2,500 to 3,000 kw., that is to increase the load factor from 50 to 60 per cent. The effect of this increase in production is to increase the cost of production in the case of the steam plant.

To	At 60 per cent Load Factor	$204,516
From	At 50 per cent Load Factor	185,990
	Increment Cost	$18,526
	Average per kw-hr.	$0.423

Or in the case of the hydro plant,

To	At 60 per cent Load Factor	$136,460
From	At 50 per cent Load Factor	132,920
	Increment Cost	$3,540
	Average per kw-hr.	$0.081

These costs, it will be noted, are very much less than the costs arrived at for a corresponding amount of primary power. The cost of secondary steam power, though much greater than that of secondary hydro power, is considerably less than that of primary hydro power.

Combination of Water and Steam Power

Such findings lead to the generalization that, in many circumstances, the economic solution of a power supply problem lies in the judicious combination of water and steam power, rather than an exclusive reliance on either one or the other.

It has been shown by typical examples, that in the regions of Canada in which water power is available it will generally prove more economical than steam-generated power. It does not follow that steam power may not have a very useful function in those same localities.

In the natural course of things it happens that water power sites offer a certain definite possibility of power generation limited in two ways,—(1) as to the total amount of energy available in a year; and (2) as to the rate at which power can be continuously generated at the season of minimum flow. It rarely happens that both or even one of these coincide with the demand for power which it is desired to meet. Either the power available may be insufficient for the needs, in which case some source of supplementary power, usually from a coal-fired steam plant, must be provided. Or, on the other hand, the water power considered may be capable of a developed capacity far beyond the immediate needs; and the extent of permanent works necessary for any development may increase the capital investment out of all proportion to the results aimed at. As has been shown, this capital investment is the main controlling factor in the ultimate cost of the hydro power, and beyond a certain point it will be more economical to resort to steam-generated power for the whole operation.

In the typical 5,000 kw. installation, suppose that the daily load curve in the winter months is as represented in Fig. 4. This shows a peak load of 5,000 kw. at 16.30 in the afternoon, with an average load of 3,000 kw. equivalent to a load factor of 60 per cent. Some such daily load factor at the time of heaviest demand would correspond to the annual load factor of 50 per cent as previously assumed.

This winter daily load occurs in the period when the water flow is usually at a minimum, which we will assume is just sufficient to generate 2,500 kw. continuously throughout the day. If this condition extends over a period of three months, during this period the hydro plant would be capable of carrying the load as illustrated between the hours

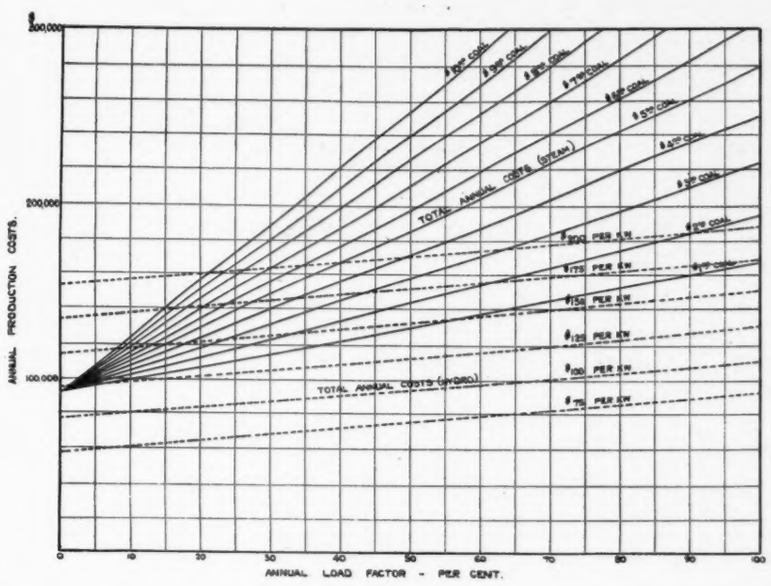
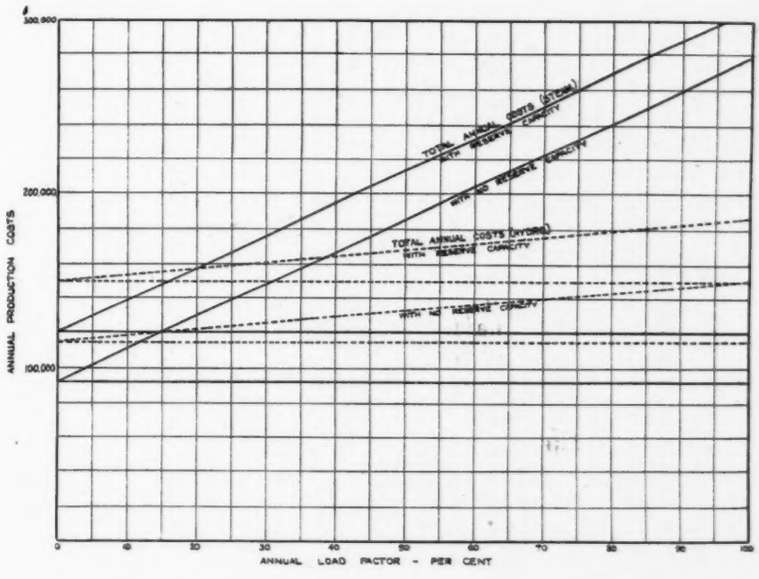


Fig. 2—Comparative annual production costs, steam and hydro 5000 kw. installation.

Fig. 3—Annual production costs, steam and hydro 5000 kw. installation with reserve for security against interruption.



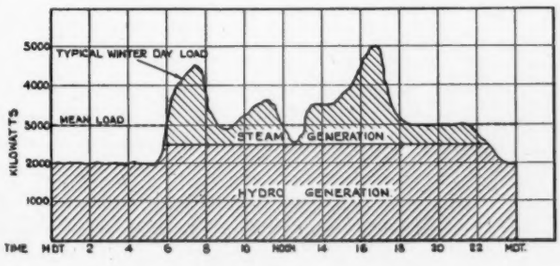


Fig. 4—Combination of hydro and steam generation.

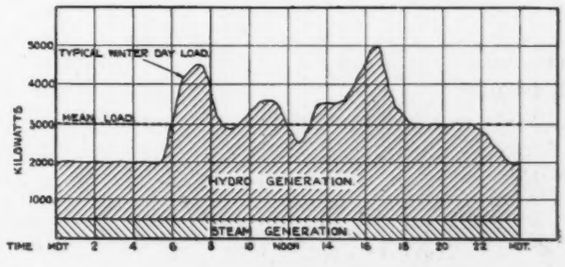


Fig. 5—Combination of hydro and steam generation.

of 22.30 and 5.45 the following morning. For the remainder of the day a supplementary steam plant would have to carry the balance of the load, averaging 716 kw. from 5.45 to 22.30 and reaching a maximum peak at 16.30 of 2,500 kw., a total output during the 24 hr. of 12,000 kw-hr. In the three months of insufficient water power capacity, the total generation by steam would amount to 1,095,000 kw-hr. which corresponds to an annual load factor of 5 per cent. The total annual cost of the power production can then be worked out as follows, using unit production costs from Fig. 1.

By hydro: 5,000 kw. plant at 47½ per cent load factor 20,805,000 kw-hr.	$131,533	
By steam: 2,500 kw. plant at 20 per cent load factor for 3 months, or 5 per cent annual load factor—½ of $93,000....	46,500	
1,095,000 kw-hr. ⅛ of $37,000	4,625	
Total production cost	$182,658	
Average cost per kw-hr.		$0.834

This production cost is inevitably much increased over the cost—$132,920 if the whole demand could be supplied by water power, but it is still somewhat lower than the cost—$185,990 of meeting the same demand by means of steam power alone.

Another method of effecting the combination of steam and hydro generation offers itself in the case where the hydro plant has sufficient head poundage to cover the fluctuations in the day's demand. This consists in arranging the supplementary steam plant to take care of the deficit in the daily output as a base load; that is, in the case under consideration, under a steady load of 500 kw. as shown in Fig. 5. The advantage of this arrangement is that, no greater load than this having to be carried by the steam plant, its capacity can be limited to this amount, which will then be operated at 100 per cent daily load factor. The hydro plant was equipped with machine capacity to deliver up to 5,000 kw. and this capacity is available at any time by drawing upon the water stored in the head pond. As a concrete example of the pondage necessary to provide for the accumulated surplus of output, which in this case amounts to 7,500 kw-hr. with a normal working head of 100 ft., 82.5 acre-feet would be required, or about 8 acres with a maximum drawdown of 10 ft. This is a condition which should easily be met in an ordinary installation.

Then the total annual cost of the same power output becomes:

By hydro: 5,000 kw. plant at 47½ per cent load factor 20,805,000 kw-hr.	$131,533	
By steam: 500 kw. plant at 100 per cent load factor for 3 months 1,095,000 kw-hr.		
1/10 of $93,000	9,300	
1/40 of $185,263	4,632	
Total production cost	145,465	
Average cost per kw-hr.		$0.664

It is not strictly true to assume that the cost of 500 kw. steam plant is proportional to that of one of 5,000 kw. or that the labor and fuel costs for operation will be in the same ratio. However it is evident that a marked economy in fuel would result from operating a small plant at constant load over a definite period, in place of operating the larger plant under a load varying from zero to maximum capacity.

Auxiliary Steam Power

The study of the foregoing combinations indicates where the use of steam power as an auxiliary to hydro-electric power may prove very useful, even in localities where there is no lack of water power.

Hydro-electric power must logically be developed in well-defined blocks, depending on the amount of flow and the head available at the site under consideration. In developing such a site, there is usually a certain amount of construction in the way of permanent works, which cannot be avoided whether the power resources are completely or only partially developed. For instance if a dam has to be constructed of a certain height to utilize the full power available, and a less height would suffice to afford the power immediately demanded, the costly part of the work is in the foundations and lower section; and the saving effected by leaving off an upper section would be comparatively negligible. It would obviously be poor policy to design the dam simply to provide for the immediate needs, as this would preclude future extension to meet increasing demands except at enormous expense. Hence the initial cost of a hydro-electric development depends in the main rather on the amount of power it is possible to develop, than on the demand for which it is desired to provide.

Carrying further the examination of the typical case earlier used for illustration, let us suppose that a survey indicates an existing demand reaching at the point of maximum peak 7,000 kw., or some 700 to 1,000 kw. more than could be relied on from the proposed hydro plant at all times. Let us suppose that further investigation discloses that a second hydro-electric site could be developed giving another 6,000 kw. output of power at similar cost. The development of the two sites would afford all the power demanded with a large surplus; but the capital and operating charges on the second plant would be almost as great as those on the first, and as a return a comparatively small amount of energy would be utilized so that the average cost per kw-hr. would be correspondingly high. The actual annual charges on the second plant might be $110,000, making the total cost of generation $242,000.

By installing a steam plant to take care of the surplus power beyond the capacity of the first plant, the capacity of that plant could be limited to the actual deficiency of the water power plant at the time of maximum demand, or say 1,000 kw. to allow a small margin. The cost of generation with this plant would include fixed charges approximately one-fifth of those determined for the typical

5,000 kw. plant, or $18,000. In addition to this there would be the operation charges. The hydro plant, capable of supplying the load demand corresponding to a peak of over 6,000 kw., would in times of normal flow be capable of carrying the full load, so that the steam plant would only need to be called on during the period of low flow or say three months of the year. The steam-plant load factor would be low, advantage being taken of the maximum output possible from the hydro plant depending on the water flow. Probably the actual load factor need not exceed 25 per cent during the three months' period of operation, which would make the annual load factor about 6 per cent. In the 5,000 kw. plant the operation cost corresponding to this load factor was $10,000; on a 1,000 kw. plant, the corresponding cost would be $2,000 plus a charge for labor which would not be reduced in proportion to the plant capacity or the load factor; also the fuel efficiency would be lower due to the intermittent nature of the operation. Probably the operation of the small plant for three months would entail an expense of $10,000 over and above the capital charge, making a total production cost of $28,000.

This it will be noted is very much less than the charge imposed by the installation of the second hydro plant.

Furthermore the steam plant would be capable of being extended by steps as the need for further supplementary power became manifest. Under the cost ratio assumed in the typical examples the economical limit of this procedure would probably be reached when the steam plant became of approximately half the capacity of the proposed hydro-electric extension. Anticipating this point it would become more economical to proceed with the second hydro-electric plant, on completion of which the steam plant would become idle.

In the further development of the hydro system to meet increasing system demand, the time will arrive when the two hydro plants become inadequate. Then the steam plant may again be pressed into service, furnishing surplus power until a further hydro plant can be economically justified.

Thus in a normally growing system mainly served by hydro-electric development, the steam plant is capable of performing a very useful function in acting as a stop-gap between progressive stages of hydro-development.

Auxiliary Hydro Power

It is quite conceivable that the converse arrangement might be profitably adopted in a limited number of situations where steam generated from coal is the primary source of power, but where a relatively small amount of water power, accompanied by reasonable pondage possibilities, is also available. In this case the scheme of operation would be to install machine capacity in the hydro plant so as to carry as much of the peak loads on hydro power as the pondage would permit. Besides any direct saving in fuel corresponding to the energy so generated, there would be the important advantages of a saving in the steam plant capacity necessary to cope with peak loads, and an increase in operating efficiency due to being able to operate the steam plant under more nearly uniform conditions.

An interesting and valuable combination of hydro and steam power can sometimes be made by utilization of off-peak power for the production of steam by electric boilers, with, possibly, storage of steam so produced to carry a short peak load, or the use of off-peak power to generate steam or heat during load curve valleys in conjunctions with fuel-fired boilers during peak periods. Such opportunities may occur in the case of purchased or available blocks of hydro power.

Many such combinations, both for the economic production of power or in connection with some heating or industrial process are found in practice, but each application must be treated as a special case.

Storage of Water

Reference has been made in the foregoing discussion to the question of storage of water, and this is a necessary adjunct in a greater or less degree of every hydro-electric development.

In every hydro plant the installation of the head works has the effect of creating a certain amount of storage in the shape of a head pond. The amount of storage so provided is usually limited and is generally known as pondage. It is usually sufficiently large, that ordinary fluctuation in demand of daily operation can be taken care of without reference to the momentary stream flow and without seriously affecting the head of water available.

Storage is the term usually applied to the more extensive provision of reservoir capacity to offset seasonal deficiencies of stream flow. Such storage is obtained usually by taking advantage of existing natural lakes, often supplemented by artificial works designed to augment the storage capacity.

The storage of potential energy in impounded water is analogous to the storage of a supply of coal convenient to a steam plant in that either places the generating plant in a position to furnish any amount of power within its capacity at call. There is, however, a fundamental difference in the manner of providing the reserve of storage. In the case of the steam plant, fresh supplies of fuel have to be purchased each season, and the provision of the storage resolves into a matter of annual cost. In the case of water storage, on the other hand, it is a matter of an initial capital outlay to provide the storage reservoir, which once completed replenishes itself year by year without any further expense. The energy thus made available, however, has a very definite cost consisting of the annual carrying charges on the initial outlay; and these may be reduced to a pro rata charge on each kilowatt hour made available by the storage.

It is obviously impossible to state any definite figure which will represent the cost of kilowatt hours so reserved in the form of water held back which would otherwise run to waste. It depends

entirely on the topography and natural features of the country which can be taken advantage of. Generally speaking, the cost of a dam with its appurtenances will increase as the cube of its height; but the higher the level of the impounded water is raised the greater will be the flooded area. The cost of creating storage tends to increase faster than the volume of water impounded; and there is an economic limit beyond which it does not pay to go. The energy represented in the water depends on the volume impounded and the total average head in the plant or plants in which the water is utilized.

As illustrating the relation between the value of stored water and coal, it may be pointed out that one acre-foot of water utilized in a plant under a head of 100 ft. would afford some 65 kw-hr. of energy at the center of distribution. The same amount of energy would be furnished by the combustion of about 81 lb. of 13,000 B.t.u. coal.

The ideal condition is reached when a sufficient amount of the run-off coming from a drainage area can be held in lakes or reservoirs, so that none of the water need be spilled or allowed to escape without delivering its quota of energy in the form of electric current. Such a condition of complete regulation is rarely capable of realization. Unfortunately in most of the northern river systems a large percentage of the annual run-off is concentrated in a short period of the year, and to impound even a substantial portion would require a greater area of territory than can be found available.

As an example, a typical year's record of the flow of the Bow River in Alberta shows that 73 per cent of the annual run-off took place during the four months from June to September. The excess over the average yearly run-off for this period amounted to some 52,000 million cu. ft., or 1,190,000 acre-feet. One study of the storage possibilities showed a total of about 250,000 acre-feet as capable of development, which it will be noted falls very far short of filling the requirements for complete regulation of the river. Even so, this 250,000 acre-feet storage, if completely utilized over the 700 ft. of total drop on this river, would make available some 159 million kw-hr. during the year. In equivalent coal consumption this would require 100,000 tons per annum. In this location the cost of coal, corrected for the equivalent of 13,000 B.t.u. heat value per pound, would be about $3.30 per ton, so that the annual cost would be $330,000. If this were set as the maximum permissible annual charge which could be justified for this output of hydro energy, and the carrying charges on the storage investment were set at 8 per cent to cover interest and almost negligible maintenance and operating charges, the justifiable investment to create the necessary storage would become $4,125,000. As a matter of fact, the cost of providing this storage was estimated at $3 per acre-foot. This was in 1914, and at this present date the estimate would probably be nearer $5. The total cost for the 250,000 acre-feet projected would therefore be of the order of $1,250,000, which is well within the limit necessary to meet the competition of coal.

In utilizing the energy of stored water in an already established hydro plant during periods when that plant would otherwise remain partially idle due to the lack of water, the production cost will not be burdened with the carrying charges of a supplementary steam plant. If, however, it becomes necessary to increase the capacity of the hydro plant in order to make use of the stored water, then further investigation becomes necessary to determine whether the carrying charges on such increased capacity, added to the cost of storage, would compare favorably with the estimated cost of production of a similar output of power and energy by means of steam.

Summary

In conclusion, it may be desirable to recapitulate the main points of the foregoing discussion.

Condensed Table of Power Production Costs

	STEAM PLANT Coal Price		HYDRO SYSTEM Installation Cost per kw.	
	$ 2	$ 8	$100	$200
	Total Annual Production Costs per kw. of maximum demand (dollars)			
For 25 per cent L. F.	24	32	17	32
For 75 per cent L. F.	34	59	21	36
	Average Cost per kw-hr. (cents)			
For 25 per cent L. F.	1.08	1.45	0.79	1.47
For 75 per cent L. F.	0.52	0.89	0.32	0.55

As between the two most widely used methods of producing electric power, steam and water power, it appears that where water power is available and can be economically developed in blocks approximately corresponding to the power demand in sight, hydro usually affords the cheaper power. This confirms the generally accepted conclusion that where water power is available, it should be exploited before resorting to fuel as a source of power.

On the other hand, where water power is not conveniently situated in relation to the center of demand, or where for various reasons the development cannot be carried out economically on a suitable scale, the use of fuel is the obvious solution.

The question of the load factor of the system to be supplied has an important bearing on the cost of the power, and also on the relative costs of steam and hydro generation.

Considering the typical medium-size developments selected for purposes of illustration, the inclusive costs of the power generated would be as shown in the following condensed table; the figures given being limited to the high and low values corresponding to high and low costs of fuel in the case of the steam generation; to high and low costs of installation in the case of the water power development; and to high and low values of the system load factor in each case.

In arriving at the figures quoted, it is assumed that good modern practice would be followed in design, construction and operation. It must be recognized that there is a continual advance in

(Continued on page 29)

Derivation and Use of Common Formulas Used in Combustion Calculations

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PART I

Formulas for Calculation of the Weight of Dry Products of Combustion

THE most common formula used for the calculation of the weight of dry products of combustion is

$$(1) \qquad \left[\frac{4\,CO_2 + O_2 + 700}{3\,(CO_2 + CO)}\right] \times C_b$$

or as sometimes written

$$(2) \qquad \left[\frac{4\,CO_2 + O_2 + 7}{3\,(CO_2 + CO)}\right] \times (C - C_r)$$

In both of these formulas CO_2, O_2 and CO are percentages by volume of these constituents in the dry gases as determined by an Orsat analysis. In the first formula the percentages are expressed as whole numbers or parts per 100 parts of sample, and in the second formula the percentages are expressed as decimals or part per part of sample. In both formulas C_b or $C - G_r$ is the weight of total carbon in the fuel, that is burned and consequently appears in the dry products of combustion, expressed as a decimal.

The first half of the formula

$$(3) \qquad \frac{4\,CO_2 + O_2 + 700}{3\,(CO_2 + CO)}$$

gives the ratio between the amount of dry gases and the amount of carbon burned. Being a ratio, it may be expressed in any weight units. Thus it may give the pounds of dry gases per pound of carbon burned, the grams of dry gases per gram of carbon burned, or the tons of dry gases per ton of carbon burned. In each case however the rest of the formula, C_b must be expressed in the same units.

These formulas are algebraic reductions of the following complete formula and are derived as shown.

$$(4) \qquad \left[\frac{44\,CO_2 + 32\,O_2 + 28\,N_2 + 28\,CO}{12\,CO_2 + 12\,CO}\right] \times C_b$$

Dividing both denominator and numerator by 4 we have

$$(5) \qquad \left[\frac{11\,CO_2 + 8\,O_2 + 7\,(N_2 + CO)}{3\,(CO_2 + CO)}\right] \times C_b$$

Since $CO_2 + O_2 + N_2 + CO = 100$, the N_2 and CO may be eliminated in the numerator.

$$(6) \qquad \left[\frac{11\,CO_2 + 8\,O_2 + 7\,(100 - CO_2 - O_2)}{3\,(CO_2 + CO)}\right] \times C_b$$

Simplifying we get

$$(7) \qquad \left[\frac{4\,CO_2 + O_2 + 700}{3\,(CO_2 + CO)}\right] \times C_b$$

Formula (4) is based on the molecular relationship between weight and volume. A molecular weight of any substance is called a mol. If expressed in pounds it is called a pound mol or simply a mol. A mol of any substance in gaseous state under the same conditions of temperature and pressure will occupy the same volume. Expressed in pounds and cubic feet this means that 44 lb. of CO_2, 32 lb. of O_2, 28 lb. of N_2 or 28 lb. of CO will occupy the same volume under the same conditions. This volume is 359 cubic feet at 32 deg. fahr. and atmospheric pressure. It follows that fractions of mols will occupy similar fractions of this standard volume or in other words, *per cent by volume is numerically the same as per cent by mol.*

Therefore if we multiply the volume fractions of a complete gas analysis by the respective molecular weights of the constituents and add the resulting weight fractions together we will have the total or molecular weight of the gas mixture and this is what is done in the numerator of formula (4).

Illustrating with a definite example of an assumed flue gas analysis,

CO_2	16.0 % by vol	$0.160 \times 44 =$	7.04
CO	0.5 % by vol	$0.005 \times 28 =$	0.14
O_2	3.0 % by vol	$0.030 \times 32 =$	0.96
N_2	80.5 % by vol	$0.805 \times 28 =$	22.54
	100.00	1.0	= 30.68

The molecular weight of the gas mixture is 30.68. Note that the molecular weight is not expressed in any weight units. It may be pounds, grams, tons or any desired unit of weight. Therefore by itself it is simply a relative weight exactly like the molecular weight of CO_2 or O_2.

To correlate this flue gas weight with the fuel, a carbon balance is made. All of the carbon burned appears in the gases as either CO_2 or CO and also, no carbon appears in the flue gases except carbon that comes from the fuel. Knowing the amount of carbon in a pound of fuel and knowing the weight of carbon in the products of combustion it is sim-

ple arithmetic to calculate the weight of the products of combustion per pound of fuel.

The denominator 12 CO_2 + 12 CO in formula (4) gives the amount of carbon in the gases. A molecule of carbon is expressed by C and a molecule of oxygen by O_2. It is evident that a molecule of CO_2 contains both a molecule of C and a molecule of O_2. Similarly a molecule of CO contains a molecule of C and a half a molecule of O_2. The molecular weight of C is 12 and the molecular weight of O_2 is 32 and the sum of these is 44 which is the molecular weight of CO_2. If therefore we have 16.0 per cent by volume of CO_2 in the gases and 0.5 per cent of CO we have 0.160 of a mol of C and 0.005 of a mol of C_2 respectively. For our purpose we are interested only in the carbon and therefore neglect the oxygen in these constituents. The total gasified carbon in the gases can be expressed in any units of weight by the following expression,

$12\ CO_2 + 12\ CO = [12 \times 0.160] + [12 \times 0.005] = 1.98$

If now we agree to express both the total dry gases, 30.68, and the total carbon, 1.98, in pounds we find that the pounds of dry gases per pound of carbon burned is

$30.68 \div 1.98 = 15.5$

and if we know that we burned 0.80 of a pound of carbon per pound of fuel, the weight of dry gases per pound of fuel is

$15.5 \times 0.80 = 12.40$ lb.

There are certain limitations to this formula that are usually negligible but should be understood so that corrections may be made if necessary.

Sulphur in fuel burns to SO_2 and SO_3 and in the flue gas analysis these gases are absorbed by the caustic solution and recorded as CO_2. The molecular weights of the two gases are respectively 64 and 80, considerably higher than the molecular weight of CO_2. Therefore when the weight of the products of combustion of high sulphur coals is calculated from the above formulas the result will be low. Ordinarily the correction is small and may be neglected.

The following calculations show the error involved for a fuel containing 7 to 8 per cent by weight of sulphur. Assuming half of the sulphur to be gasified, the rest going to sulphates in the refuse, and the resulting gas to be half SO_2 and half SO_3, there will be about 0.3 to 0.5 per cent by volume of sulphur gases in the products of combustion that will be determined and recorded as CO_2 in the Orsat analysis. Using an average molecular weight of 72 for the mixed sulphur gases, the correct formula for the calculation of the dry products of combustion would be,

$$(8)\quad \left[\frac{44\ CO_2 + 72\ (SO_2 + SO_3) + 32\ O_2 + 28\ N_2 + 28\ CO}{12\ CO_2 + 12\ CO}\right] \times C_b$$

$$(6)\quad \left[\frac{4\ CO_2 + 11\ (SO_2 + SO_3) + O_2 + 700}{3\ (CO_2 + CO)}\right] \times C_b$$

The dry gases per pound of fuel for the above example, assuming the 16 per cent CO_2 to be 15.6 per cent CO_2 plus 0.4 per cent $SO_2 + SO_3$, as calculated from this formula is equal to 12.7 pounds.

The uncorrected formula would be

$$(10)\quad \left[\frac{44\ (CO_2 + SO_2 + SO_3) + 32\ O_2 + 28\ N_2 + 28\ CO}{12\ (CO_2 + CO + SO_2 + SO_3)}\right] \times C_b$$

and the dry gases calculated by this uncorrected formula is equal to 12.4 lb., about 2½ per cent low.

In blast furnaces and lime kilns where the products of combustion are mixed with products of decomposition that contain carbon, the above formula, being based on a carbon balance, cannot be used without corrections. Limestones decompose and give off CO_2 that appears in the gases but this CO_2 does not represent burned carbon from the fuel so that the carbon balance will not give the ratio between flue gases and fuel unless correction is made for the carbon from the limestone.

No such correction is necessary for carbonates present in the mineral matter of a fuel because the total carbon in the fuel as determined by an ultimate analysis includes mineral as well as organic carbon.

Likewise no corrections are necessary for fuels containing large amounts of N_2 or CO_2, such as blast furnace gas or producer gas, as long as C_b represents all of the carbon in the fuel that appears as CO_2 and CO in the gases.

A simplified form of the above formulas, convenient for quick calculations is

$$(11)\quad \left[\frac{240}{CO_2} + 1\right] \times C_b$$

This formula is developed as follows and involves two assumptions. First that the per cent of CO in the gases is negligible and second that the sum of the percentages of CO_2 and O_2 in the gases is equal to 21. The latter assumption involves no serious error when the hydrogen content of the fuel is low. With increasing hydrogen content the sum of the CO_2 and O_2 becomes increasingly less than 21 per cent because part of the oxygen goes into the water vapor produced by the combustion of the hydrogen and the error involved becomes appreciable.

$$(1)\quad \left[\frac{4\ CO_2 + O_2 + 700}{3\ (CO_2 + CO)}\right] \times C_b$$

Assuming that the CO is negligible and that the sum of the CO_2 and O_2 is equal to 21 per cent, the CO and C_2 may be eliminated.

$$(12)\quad \left[\frac{4\ CO_2 + (21 - CO_2) + 700}{3\ CO_2}\right] \times C_b$$

$$(11)\quad \left[\frac{CO_2 + 240}{CO_2}\right] \times C_b$$

The error involved in the use of this formula for calculating the products of combustion for a fuel

oil containing 11-12 per cent hydrogen is shown below. An average gas analysis of the dry products of combustion from fuel oil may be taken as

CO_2	12.8 per cent by volume
O_2	3.7 " " " "
N_2	83.5 " " " "

and the carbon burned may be taken as 0.84 lb. per lb. of fuel oil. The dry gases per pound of fuel oil as calculated by the correct method is

$$\frac{(4)\ (12.8) + 3.7 + 700}{(3)\ (12.8)} \times 0.84 = 16.51 \text{ lb.}$$

As calcuated by the special formula the result is

$$\left[\frac{12.8 + 240}{12.8}\right] \times 0.84 = 15.81 \text{ lb.}$$

which result is a little over 4 per cent low. For natural gases which contain as high as 25 per cent hydrogen the error involved is proportionately greater.

No formula should be used unless its derivation and limitations are known. It will be found that a study of the molecular relationships expressed by the elementary equations for the combustion of carbon, hydrogen and sulphur will be invaluable in understanding the common formulas and equations used for combustion calculations.

The Economics of Electric Power Production

(Continued from page 26)

what constitutes the best practice, more particularly in respect of steam generation, and consequently any figures given will become due for revision from time to time to keep pace with the advance in the art.

It cannot be too strongly stressed that the figures given in the foregoing table and in the more comprehensive diagrams from which this table is condensed, are not to be taken as of universal application. So long as the conditions surrounding the power production generally conform to those discussed in the preceding text, these figures may be taken as representing pretty closely the actual results that may be expected. They are not intended to be stretched to include special or abnormal conditions, which may give rise to very different results.

It is evident that there exist conditions in which there can be no doubt as to the most economical method of generating power. There exists however an intermediate range of conditions in which the ultimate costs balance one another so nearly, that a careful estimate of all construction costs, along with as accurate as possible a forecast of the conditions of usage of the power output, are necessary to decide which method is the more attractive. It is in dealing with such problems that the method of analysis suggested may prove of value.

Such investigations frequently lead to the conclusion that the conditions are such that the most economic results can be attained by a judicious combination of the two methods of power production. This opens up a wide field of investigation, the general nature of which is indicated by a very elementary example. It is possible to broaden and amplify this analysis to take account of all the significant factors in any particular situation. By such methods only can the true economics of power production be determined.

Sectional Meeting of World Power Conference

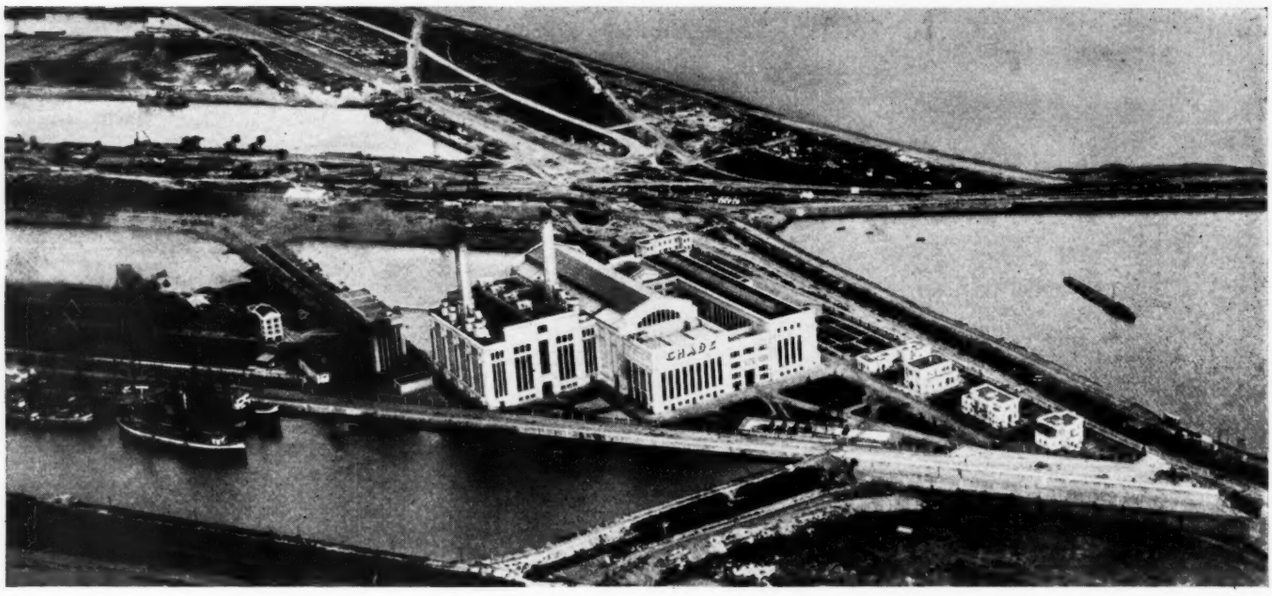
A sectional meeting of the World Power Conference is planned for the summer of next year. This meeting is to take place in Copenhagen, Stockholm and Oslo, June 26 to July 10. It will be devoted to energy requirements in large industrial plants and how they may best be met. The questions of generated vs. purchased power, generation of base load and purchase of peak power, coordination of hydraulic and fuel plants, supply of process heat requirements in various industries, electrical heating, use of power in heavy mechanical drives, energy problems in transportation for land and marine traffic will be covered. This meeting should be of great interest and productive of results in the field of industrial power, since the importance of the problems to be considered to recovery of industrial prosperity and future success of industries is eident.

The Norma-Hoffman Bearings Corp., Stamford, Conn., have appointed H. J. Ritter sales manager of the organization. Mr. Ritter is also assistant secretary of that company. Mr. Ritter brings to his new duties the experience of years as a sales engineer, later, as manager of the New York sales office.

C. H. Crocker, comptroller of the Worthington Pump and Machinery Corporation, has been elected vice-president in charge of a newly organized unit called the comptrollers' division of the American Management Association.

Sarco Company, Inc. has incorporated a new company, to be known as the Spirax Manufacturing Co., Ltd., in London, for the manufacture of Sarco traps and other specialties for the British market.

The Puerto Nuevo Station in Buenos Aires



A Description of the Most Important Power Station in South America

By
DAVID BROWNLIE, LONDON

SINCE additional boiler plant equipment is now to be installed at the new Puerto Nuevo superpower station of the Compania Hispano-Americana de Electricidad in Buenos Aires, it will be of interest to give a description of this important plant, designed for an ultimate capacity of 630,000 kw.

The first section of 105,000 kw. went into operation in 1929, with 8 steam boilers and reheaters and 2 turbine-generators each of 52,500 kw. normal duty. The initial installation was characterized by medium-size boilers, moderate steam pressure and temperature, pulverized fuel firing of the storage type with separate pulverizer house, centrifugal dust separators, and reheating between the stages of the turbine. There is now on order 2 more boilers, as an addition to the first section, which will be erected towards the end of the present year.

The site of the station is on the bank of the River Platte, at the extreme point of the new port (Puerto Nuevo) of Buenos Aires. It is particularly well situated as regards facilities for bringing coal by sea and for insuring an unlimited supply of cooling water. The total area of the site is about 44,478 acres which allowed ample space for future extensions.

The initial contracts for the station were placed in the early part of 1926, and on January 25, 1927, excavations for the foundations were begun. On September 15, 1927, work started on the boilers, and on January 25, 1928, the erection of the turbines was started. The first 52,500 kw. turbo-generator was put into service on December 29, 1928, and the completed first section of the station was started up in 1929.

The total weight of the steelwork used in the construction of the buildings was about 10,000 metric tons (2204 lb.) not including 2,400 metric tons for the coal handling plant, while the total area of the buildings including staff dwellings, offices, stores and other auxiliary buildings is 172,224 sq. ft.

With regard to the coal used at the Puerto Nuevo Station this is of the most varied character, the qualities specified being as follows, which incidentally is a striking example of one of the main advantages of pulzerized fuel firing, that of

being able to operate at the highest efficiency, almost irrespective of the quality of the fuel:—

Name of Coal	Volatile Matter per cent	Ash per cent	Heating Value * B.t.u. per lb.
(1) Ruhr coal (Germany)...	20–24	9	13320
(2) Cardiff (S. Wales).....	12–13	6-7	13680–13860
(3) Mixed Ruhr and Cardiff coal (a)	20–22	7-9	13680
(4) Mixed Ruhr and Cardiff coal (b)	14	....	12060
(5) Pocahontas and New River slack	16.74	9.10	13860
(6) Pocahontas and New River run of mine.....	17.66	4.62	14580
(7) Pocahontas and New River washed smalls...	16.00	9.70	14292
(8) Belgian mixed coals (Hensies-Pommeroeul district)	17–22	6.70	14580

* B.t.u. for coals Nos. 1 to 4 are given as the lower heating value while those for coals 5 to 8 are the higher heating value. The sulphur content of the Pocahontas and New River coal is within the range of 0.59 to 0.75 per cent, with 0.60 per cent for the Belgian mixed coals.

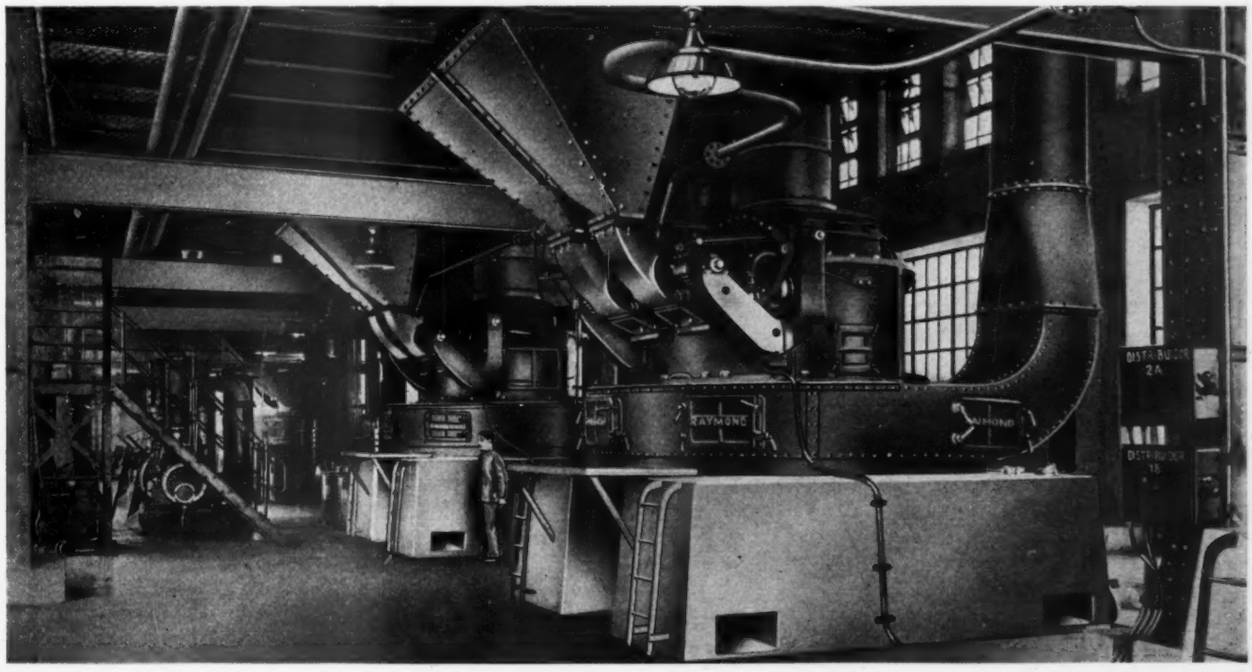
All the coal arrives by boat and is removed from the holds by two traveling cranes, each of which is able to handle 100 to 115 metric tons per hr., these grabs having a capacity of 141 to 176 cu. ft. The maximum height of each of the traveling cranes is 138 ft. while the weight is approximately 315 metric tons.

The coal is taken by belt conveyors to the crushing plant of the pulverizer house, which is 154 ft. long by 49 ft. wide by 82 ft. high, and has four overhead coal bunkers of 750 metric tons total capacity. Also there is a storage dump having an ultimate capacity of 40,000 metric tons, which is served by two traveling, overhead, grab-bucket cranes, each of which has a capacity of 300 metric tons per hour.

As indicated by the photographs, this Puerto Nuevo station is of fine appearance, and in this connection it is interesting to note that the coal storage dump has trees planted around it to hide the coal from view.

The coal is crushed so that it will all pass through a 0.79 in. ring, and then discharged from the pulverizer house hoppers through automatic weighing machines to steam dryers to reduce the moisture. It is then pulverized in four Raymond 6-roll super-mills, each of which has a capacity of 15 metric tons of coal per hour, or a maximum capacity of 1440 metric tons of coal per 24 hr. As a rule, however, three mills only are working for 20 hr. out of the 24, equivalent to 900 tons per 24 hr.

Each of these mills is driven by a 200 hp. motor within a speed range of 485 to 500 r.p.m. and is equipped with duplex coal feeders driven by small variable speed motors of 1 hp. each, running at 950 r.p.m. There is a cyclone separator and exhauster of the No. 14 Raymond type for each mill with 52 in. dia. impellers, running at 960 r.p.m., drawing the air from the cyclone and discharging it to the base of the mill. The cyclone separator discharges through a bifurcated chute to two screw conveyors which take the pulverized coal to storage bunkers, two in number, each of 50 metric tons capacity, divided vertically into two compartments. These pulverized coal storage bunkers are situated between the third and fourth mills, and there is included a duplicate set of screw conveyors so that the discharge of pulverized coal from any of the four mills can be conveyed to either of the two compartments. The normal capacity of each of the screw conveyors is 45 tons of pulverized coal per hr., that is equal to the output of the three mills representing normal running.



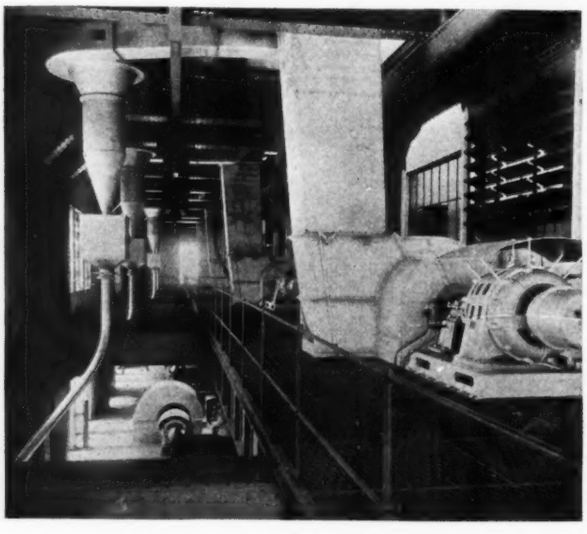
Pulverizer plant, Puerto Nuevo Station.

The coal dryers, of which there are two for each pulverizing mill, are of the vertical, wide-diameter, cylindrical type with horizontal steam-heated floors and slow-moving rabble arms, driven from vertical shafts, the coal being scraped round each of the heated floors in turn and dropped through from one to the other. At the same time hot air from a high-pressure steam-operated heater is passed upwards through the heater. The normal capacity of each of the dryers is 7½ metric tons of coal per hour, each mill being supplied by two dryers. Each dryer is driven by a variable speed 3 hp. motor running within a speed range of 400 to 1200 r.p.m.

The fans for circulating the hot air have a maximum duty of 7000 cu. ft. of air per min. at 150 deg. fahr. operating at 2 in. w.g. and driven by a 5 hp. motor running at 950 r.p.m. The air and steam are discharged to the atmosphere through small cyclone separators having water sprays to prevent dust nuisance.

Pulverized coal is fed from the storage bunkers in the pulverizer house to the receiving bunkers in the boiler house by two sets of Fuller-Kinyon pumps, one of which is a standby. The capacity of each of these pumps is 75 metric tons of coal per hour, that is equal to the full capacity of 5 pulverizing mills of 15 tons of coal per hour each, for which number of mills the present pulverizer house is built. Each of these pumping sets has a 2-cylinder, vertical air compressor having a capacity of about 700 cu. ft. of air per min., working at 30 to 100 lb. per sq. in. pressure and 760 r.p.m. and driven by a 150 hp. electric motor. The whole of the pulverizer transport pipe line for the eight boilers is 8 in. dia. and approximately 1200 ft. long.

In the present completed plant there are six boilers of the B. & W. cross-drum marine type, with two reheater units of the same make. All boilers are fired by the Lopulco Storage System of which the Raymond mills, already described, are a part. Each boiler has a heating surface of 15,400 sq. ft. and is designed for a continuous maximum evaporation of 206,000 lb. of water per hr., with a peak capacity of 226,600 lb. per hr. for short periods. The most economical rate of operation is 144,000 lb. per hr. The steam pressure is 585 lb. per sq. in., and the superheated steam temperature 770 deg. fahr. Each boiler is equipped with an economizer of 11,054 sq. ft. and a Usco plate type air heater of 27,000 sq. ft.



Induced draft fans and dust collectors.

As regards fans, each boiler is provided with a Sirocco double-inlet, induced draft fan, 64 in. dia., with a maximum capacity of 140,000 cu. ft. of gas per min. at 375 deg. fahr. These fans are operated at 7¾ in. w.g., quite a moderate figure in comparison with the latest installations which are characterized by suctions of from 10 to 12 in. w.g. while even up to 16 in. is being considered. The fans are driven by two independent, variable-speed motors, the largest of which is 350 hp. but capable of being operated down to about 100 hp. The other motor is of smaller size, giving from 22 to 190 hp. within a speed range of 280 to 575 r.p.m., the object of this combination being to allow the fans to be run within a range of from 30 per cent of the normal load to the full overload.

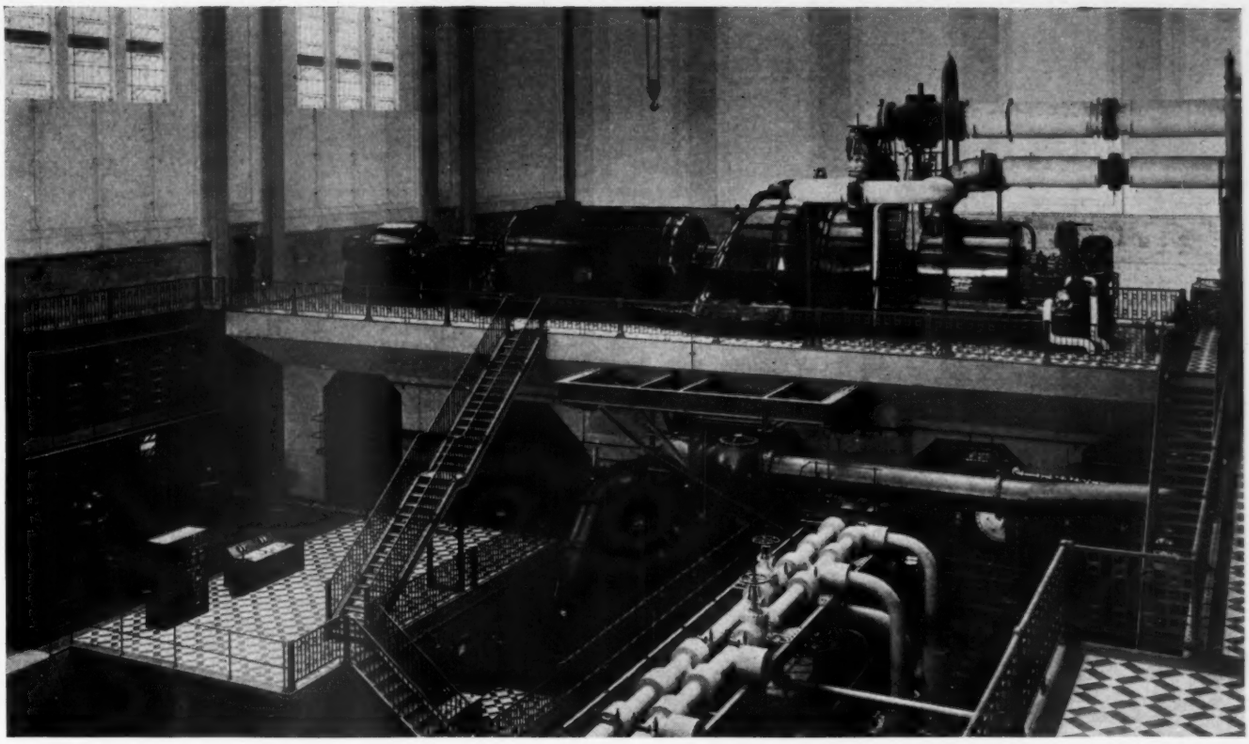
The induced draft fans each discharge to a Davidson "S. P." type cyclone dust separator of 9 ft. 6 in. dia., leading to two steel chimneys, each of which is 21 ft. dia. at the base and 17 ft. at the top, reaching to a height of 140 ft. above the boiler house roof or 227 ft. 6 in. above the firing floor. One chimney serves four boilers and the other, two boilers and two reheaters, the arrangement of the boiler house being one row of four boilers, and opposite it, the other two boilers and two reheaters.

While full performance details are not available it is stated that these collectors are giving satisfactory results, and that the average screen analysis of the dust being separated is 99.8 per cent through a 40-mesh screen, 95.78 per cent through a 100-mesh screen, and 52.66 per cent through a 200-mesh screen.

As regards forced draft, each boiler has one Sirocco fan with a maximum capacity of 74,000 cu. ft. of air per min. at 4 in. w.g., being of the double-inlet type 47½ in. dia. and also being driven by variable-speed motors in duplicate, one of which runs at 425 to 575 r.p.m., giving 45 to 110 hp., and the other at 85 to 280 r.p.m., giving 13 to 64 hp.

The two reheater units have a total heating surface of 3940 sq. ft. with three drums, and include economizer, air heater, and fans on the same general lines as the boilers. Normal maximum duty of each of these reheaters is the raising of approximately 532,400 lb. of steam per hr. at 125 lb. pressure and 485 deg. fahr. temperature to a temperature of 770 deg. fahr. For short periods, as much as 595,000 lb. of steam per hr. can be handled by each unit, while the most economical rate of working, is 422,400 lb. of steam at 100 lb. pressure raised from 433 deg. fahr. to 770 deg. fahr. It is possible to operate with as low as 228,000 lb. of steam per hr.

The furnaces of both the boilers and reheater



Turbine house showing one of 52,500 kw. units with auxiliaries.

units are of Lopulco design, with rear walls and part of each of the side walls composed of water-cooled surfaces of the bare-tube type. The other half of the side walls, however is of firebrick with plain water-cooled steel tubes, so arranged as to reduce the cooling surface in the immediate neighborhood of the burners. This arrangement was employed because of the very wide range of duty required. Further, the front wall of the combustion chamber is of the air-cooled refractory type with hot air from the air heaters passing through the wall ducts and into the furnace, entering at right angles to the flames from the burners so as to give a turbulent and mixing effect. The volume of each furnace is 25,000 cu. ft., with a maximum height under the boilers of 31 ft. Each boiler has 10 vertical Lopulco burners of the stream-line type installed in the arch above the front wall of the combustion chamber.

All furnaces are equipped with the usual water screens while the ash handling is on the Usco water sluice principle, the present installation including two 9 in. water circulating pumps with a capacity of 2000 gal. per min., driven by 35 hp. motors running at 720 r.p.m.; one of these sets is a standby.

The feeders are of the variable speed, Lopulco duplex type, one feeder supplying two burners, or a total of five feeders for each boiler. The primary air which enters the burners with the coal is supplied by separate forced draft fans running at 16 in. w.g. and driven by a 60 hp. motor.

The extension now under way is to consist of 2 boilers with economizers, air heaters, fans and dust collectors, and other equipment. They will be exact duplicates of the present 6 boilers. For this purpose the boiler house is being extended and one new boiler placed at the end of each of the existing row of boilers and reheaters. The new units will be served by the existing chimneys.

The present turbine house is 341 ft. long, 118 ft. wide and 154 ft. high, being 131 ft. above ground level. It is designed for four turbo-generators each of 52,500 kw. normal capacity. At the present time two of these turbo-generators are erected and running, that is Nos. 1 and 4. Eventually it is the intention to extend the turbine house to a total length of 820 ft. which will permit the housing of twelve 52,500 kw. capacity turbo-generators, to make a total of 630,000 kw. on maximum normal duty with all the turbines running. Turbine house equipment includes a large overhead traveling crane, with a span of 82 ft. having two hooks for lifting 20 to 120 metric tons.

The present turbo-generators were supplied by the General Electric Company, being 2-stage turbines direct coupled to alternators running at 1500 r.p.m. for 52,500 kw. normal and 54,350 kw. maximum. These are the largest turbo-generator sets in operation on the South American Continent, each weighing, with all the accessories, including condensers and pumps, approximately 600 metric tons, and occupying a space of 7750 sq. ft. The condensers have an area of 53,820 sq. ft.

In writing this description I have to thank sincerely the Compania Hispano-Americana de Electricidad for supplying photographs and full information, and also International Combustion, Ltd., of London, and Davidson & Co., Ltd., of Belfast.

NEW CATALOGS AND BULLETINS

Any of the following publications will be sent to you upon request. Address your request direct to the manufacturer and mention COMBUSTION Magazine

Boilers

Catalog B-32-3 issued by Foster Wheeler Corporation, describes the several types of boilers manufactured in the corporation's shops. The first portion of this catalog is devoted to a description of sectional header boilers for stationary plants. The next part is devoted to bent-tube boilers for stationary plants. Photographs and drawings show different arrangements of bent-tube boilers including three-drum, four-drum and six-drum types. The third section describes boilers of smaller size including the three-drum, bent-tube, low-head type and various horizontal fire-tube types. 48 pages, 8½ x 11—Foster Wheeler Corporation, 165 Broadway, New York.

Boiler Baffle Walls

A folder entitled The Beco-Turner Baffle Wall has just been issued by the Boiler Engineering Company. This folder describes the Beco-Turner baffle and how it should be applied. It includes numerous examples of how boiler efficiency and capacity have been improved by proper baffling. Photographs of actual installations on all types of water tube boilers are reproduced. 6 pages, 8½ x 11—Boiler Engineering Co., 744 Broad St., Newark, N. J.

Combined Barometer and Vacuum Recorder

Bulletin No. 150 issued by the Uehling Instrument Company sets forth the construction and operation details of the Uehling Combined Barometer and Vacuum Recorder. This recorder is designed to determine, A, the absolute back pressure in steam turbine and condensing plants, B, the barometric pressure, C, the condenser vacuum, D, the existence of air leakage into the condenser and its connections, and, E, the ability of the condenser to handle the load. This bulletin is illustrated and also contains tables and charts. 6 pages, 8½ x 11—Uehling Instrument Company, 473 Getty Avenue, Paterson, New Jersey.

Constant Pressure Regulators

Bulletin CP-3, recently issued by the Brooke Engineering Company describes the Brooke Regulators. These regulators are especially designed for the operation of balanced Port Piston Valves to maintain constant reduced pressures, regardless of the valve opening required. The motive power in the regulator is supplied by a motor small enough to permit connecting the regulator to any 110 volt A.C. or D.C. lighting circuit. Line drawings illustrate the various details of construction. 4 pages, 8½ x 11—Brooke Engineering Company, Inc., 3640 N. Lawrence Street, Philadelphia, Pennsylvania.

Dust Handling Equipment

A catalog entitled Hydrovac Dust Handling has recently been issued. The Hydrovac System is a method of transporting finely divided particles such as fly dust and siftings without the escape of any particles. 8 pages, 8½ x 11—The Allen-Sherman-Hoff Company, Philadelphia, Pa.

Flow Meters

Cochrane High Torque Flow Meters is the name of a publication recently issued by Cochrane Corporation. The principle on which this meter operates is described in detail. Photographic reproductions, sketches and tables are included. 32 pages, 8½ x 11—Cochrane Corporation, 17th Street and Allegheny Avenue, Philadelphia, Pa.

Heat Exchangers

A new catalog recently issued by the Henry Vogt Machine Company describes the Vogt Heat Exchangers. These exchangers are designed for every temperature, pressure and vacuum service. They are built of high tensile strength cast iron or of welded steel. 8 pages, 8½ x 11—Henry Vogt Machine Co., Louisville, Ky.

Indicating and Recording Gages

A new catalog has been issued by the Hays Corporation describing the various types of gages manufactured by them for measuring and recording draft, pressure, differentials and temperature. It includes photographic reproductions of the various types of gages, and also charts and tables. 20 pages, 8½ x 11—The Hays Corporation, East 8th Street, Michigan City, Indiana.

Oil Burners

The Hammel Oil Burner is the title of a pamphlet just published by the Hammel Oil Burner Company. The improved Hammel Patented Oil Burner has been designed to burn hydrocarbon fuel of any gravity without using more than a small fraction (1 per cent to 2 per cent) of the steam for atomization. The oil does not have to be preheated. This burner has been applied to stationary steam boilers, locomotives, oil stills, etc. Photographic reproductions, line drawings and a table of prices are included. 8 pages, 8½ x 11—Hammel Oil Burner Company, 724 W. Vernon Avenue, Los Angeles, California.

Roller Bearings

A book splendidly arranged and printed, has just been issued by The Timken Roller Bearing Company. In this publication is set forth the early history of the company and particularly the development of the design of the Timken Roller Bearing. This book is divided into two parts bearing the following headings: Part one, Correct Bearing Design Material and Construction, Part two, Experience in Applying the Bearing to Meet the Individual Specialized Problems. 28 pages and cover, 8½ x 11—The Timken Roller Bearing Company, Canton, Ohio.

Stainless and Heat Resisting Pipe and Tubes

A book has recently been published entitled National USS Stainless and Heat Resisting Pipe and Tubes. The National Alloy Tubular Products are described and their uses and applications indicated. The physical and chemical properties of the alloys are shown in tables and charts. 40 pages and cover, 8½ x 11—National Tube Company, Pittsburgh, Pa.

Steam Jet Ejectors

Bulletin G-1 entitled Steam Jet Ejectors has been issued by the Elliott Company. The theory of ejectors is first defined, then the Elliott single-stage, two-stage and three-stage ejectors are described together with special ejectors and thermo-compressors. The bulletin is fully illustrated with photographic reproductions and line drawings. Facts concerning the operation and application of ejectors and a table showing materials of construction are included. 24 pages, 8½ x 11—Elliott Company, Pittsburgh, Pennsylvania.

Steam Plant Saves $33,000 a Year

A reprint issued by Combustion Engineering Corporation entitled "Coal-Fired Industrial Steam Plant Saves $33,000 a Year," describes how savings were effected by replacing four horizontal return tubular boilers with two Heine Boilers fired by Coxe Traveling Grate Stokers. 6 pages, 8½ x 11—Combustion Engineering Corporation, 200 Madison Avenue, New York.

NOTICE

Manufacturers are requested to send copies of their new catalogs and bulletins for review on this page. Address copies of your new literature to

COMBUSTION
200 Madison Ave., New York

NEW EQUIPMENT

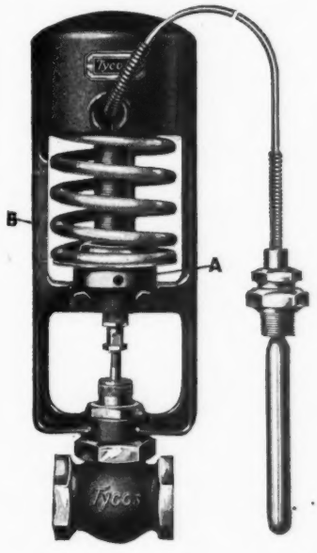
of interest to steam plant Engineers

Self-Acting Temperature Regulator

The Taylor Instrument Companies, Rochester, N. Y., have developed a new Tycos self-acting temperature regulator—of surprising accuracy and sensitivity.

The most important improvement in this regulator is the practical elimination of friction, making its over-all performance vastly superior.

Friction seriously interferes with the operation of most regulators of the self-



acting type, manifesting itself by the inability of the valve stem to assume the same position for the same bulb temperature. Actually what occurs is that the bulb temperature must be changed several degrees before the valve stem starts to move.

Obviously during this period when the temperature of the bulb is increasing or decreasing with no resultant stem movement, the disc in the valve does not move and consequently the rate of flow through the valve is not changed. This range can be called the "dead period." In some regulators, it amounts to over 10 deg. fahr. In the new Tycos regulators it has been reduced until practically eliminated and the stem moves almost simultaneously with the temperature change, giving a throttling control that is very desirable. This practical elimination of friction is a major improvement, contributing to accurate performance.

Another element which contributes greatly to the superior performance of the Tycos regulator and its adaptability to the greatest variety of applications is the construction of the tube system. This consists of a double seamless bellows, extra heavy-walled copper connection tubing and stainless steel bulb of exceptionally small size and thin wall.

This tube system requires a minimum volume of actuating medium, which makes a small bulb possible and together with the thin wall of the bulb and its shape results in a tube system that responds quickly to temperature changes. Of great value also is the overheating features of the tube system which permits heating the bulb beyond the upper limit of the range of the instrument without injury.

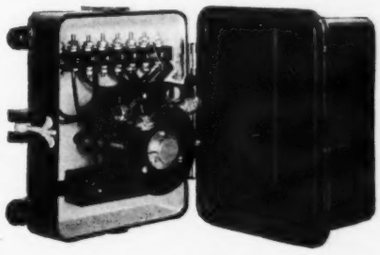
Throughout the construction and design of the Tycos regulator the essential elements which insure an instrument of superior performance—accuracy, quick response and dependability—have been developed to a high degree.

Motor Operated Relay

The Automatic Temperature Control Company, 34 East Logan Street, Philadelphia, Pa., has developed the Motomerco relay which is equipped with motor operated mercury contact tubes and is mounted as standard in a cast aluminum case with a hinged gasketed cover.

The Motomerco relay was designed primarily to select one of three control circuits when actuated by an instrument having sustained contacts for low and high positions only with an intermediate no-contact position. Preferably the instrument low and high contacts should be individually adjustable to permit change in width of the intermediate no-contact or holding position to meet local conditions. The design of the relay is such that it will automatically return to its mid-position and make a control circuit when the actuating instrument is at its intermediate no-contact position.

By means of the Motomerco relay, it is possible to use an instrument with a single sustained contact such as a thermometer, pressure gage, etc., for operating two separate control circuits as required to function a two-position motor operated valve, thus eliminating the customary low position contact so that there is no interference with free movement of the in-



strument stylus and its true reading when conditions are such that the stylus is below the control setting.

Where the Motomerco relay is to be used to operate two separate control circuits from an instrument with a single sustained contact, a simple mechanical adjustment can be provided to introduce any desired time lag up to 25 sec. thus delaying a change in control circuits should the actuating instrument make an accidental contact.

Where it is desired to use three colored signal lights to indicate the condition of temperature, pressure or other controllable factor and the actuating instrument has intermittent instead of sustained contacts for low, normal and high positions respectively, the Motomerco relay can be wired as a three-position holding-in type so that it will remain in the position last energized. This will allow it to hold steadily the signal light corresponding to position of the instrument stylus, without the objectionable off-and-on action which would otherwise be the case.

There are no open contacts with the Motomerco Relay as all circuits are broken by mercury enclosed in glass tubes sealed under gas pressure. The actuating instrument will not develop contact trouble, even when subjected to severe vibration, as the motor used in the Motomerco relay draws but 4 watts at 110 volts, A.C.

Automatic Combustion Indicator-Recorder

The C. J. Tagliabue Mfg. Co. of Brooklyn, N. Y., manufacturer of recording and controlling instruments are now offering a new all metal automatic flue gas analyzer known as the Tag-Mono Indicator-Recorder.

The instrument not only indicates and records the per cent of CO_2 but also gives an accurate record of the per cent of combustibles, such as $CO+H_2$ which may

be present in the flue gases. This is conceded to be extremely important in getting the whole story of what is taking place in the furnace.

This new Tag Development which operates on the orsat principle, analyzes chemically and indicates and records electrically. The large illuminated dial and pointers can be easily seen at a distance of 30 ft., and the records are made on a 9 in., 24 hr. circular chart with open graduations so as to invite a precise study of the continuous performance curve.

A new catalog No. 1025 fully describes and illustrates the new Tag-Mono Indicator Recorder and may be had by writing the company.

American Standards Association to Develop Specifications for Clean Bituminous Coal

The American Standards Association, at the request of the American Institute of Mining Engineers, is undertaking the development of specifications for clean bituminous coal. The project was originally proposed by the Institute in May, 1931, and has been the subject of an extensive canvass of producing and consuming groups during the past year. Opinion has now crystallized into the view that, although the project is difficult, attempts should be made to harmonize the divergent interests affected by such a standard. For the present it is proposed that the work of the committee in charge of this work should be confined to groups located west of the Mississippi River. Although the scope as yet has not been definitely decided upon, correspondence with interested organizations indicates that the following covers, in a general way, the purpose of this project:

Specifications outlining allowable limits of inacceptable material in prepared bituminous coal, whether mechanically cleaned or not, in sizes coarser than 2 in., the purpose being to define clean coal and not to standardize screen sizes.

Clean bituminous coal is at present simply a name with no guide or limit to determine what is meant when the term is used by coal mine operators or dealers. The purchaser has no method by which to determine whether or not his preparation is satisfactory, and the distributor or retailer cannot be assured of a supply of uniform character. Claims and counter claims are constantly occurring, and, due to the lack of specifications, the task of harmonizing the varying viewpoints is a difficult one. A standard will benefit both producer and consumer—the producer by furnishing a gage covering the purchase of equipment and the degree of preparation required to provide a satisfactory product; and the consumer through the assurance of receipt of uniform coal complying with accepted specifications.

The Kron Company, Bridgeport, Conn., manufacturers of a complete line of industrial scales, announces the appointment of Winslow Sampson as their Pittsburgh District Manager, with offices at 302 Penn Ave. For many years Mr. Sampson was associated with S. K. F. Industries.

Blake R. Van Leer, assistant secretary of the American Engineering Council, has been appointed dean of the College of Engineering, University of Florida, to succeed the late John R. Benton, who died in 1930. Mr. Van Leer, who is widely known for his work in the field of hydraulics, has had twelve years of teaching experience at the University of California.

Boiler, Stoker and Pulverized Fuel Equipment Sales

As reported by equipment manufacturers to the Department of Commerce, Bureau of the Census.

Boiler Sales

Orders for 242 boilers were placed in April by 72 manufacturers

	Number	Square feet
April, 1932	242	240,902
April, 1931	631	653,115
January to April (inclusive, 1932)	967	1,057,870
Equivalent period, 1931	2,211	2,208,461
Total, 1931	7,508	6,327,262

NEW ORDERS, BY KIND, PLACED IN APRIL, 1931-1932

Kind	April, 1931 Number	April, 1931 Square feet	April, 1932 Number	April, 1932 Square feet
Stationary:				
Water tube	57	329,276	28	121,908
Horizontal return tubular	42	49,932	26	30,407
Vertical fire tube	59	13,358	23	5,345
Locomotive, not railway	24	31,090	5	5,860
Steel heating	406	200,329	146	66,109
Oil country	8	8,186	....	.....
Self contained portable	29	17,949	12	8,897
Miscellaneous	6	2,995	2	2,376
Total	631	653,115	242	240,902

Mechanical Stoker Sales

Orders for 95 stokers totaling 14,909 hp. were placed in April by 55 manufacturers

	Installed under: Fire-tube boilers No.	Fire-tube boilers Horsepower	Water-tube boilers No.	Water-tube boilers Horsepower
April, 1932	80	9,828	15	5,081
April, 1931	88	11,914	36	13,906
January to April (inclusive, 1932)	273	36,972	109	54,022
Equivalent period, 1931	367	48,160	165	58,266
Total, 1931	1,889	252,571	574	187,507

Pulverized Fuel Equipment Sales

Orders for 4 pulverizers totaling 18,200 were placed in April

STORAGE SYSTEM

	Pulverizers: Total Number	No. for new boilers, furnaces and kilns	No. for existing boilers	Total capacity lb. coal per hour for contract	Water-tube Boilers: Number	Total sq. ft. steam generating surface	Total lb. steam per hour equivalent
April, 1932	..	..	..	.....	..	.....	.....
April, 1931	2	2	..	60,000	1	34,300	592,000
January to April (inclusive, 1932)	..	..	..	.....	..	.....	.....
Equivalent period, 1931	7	6	1	220,000	3	114,577	1,671,000
Total, 1931	8	7	1	250,000	4	126,471	1,797,000

DIRECT FIRED OR UNIT SYSTEM

	Pulverizers: Total Number	No. for new boilers, furnaces and kilns	No. for existing boilers	Total capacity lb. coal per hour for contract	Water-tube Boilers: Number	Total sq. ft. steam generating surface	Total lb. steam per hour equivalent
April, 1932	4	2	2	18,200	3	16,050	179,500
April, 1931	9	8	1	49,250	6	46,300	538,200
January to April (inclusive, 1932)	25	19	6	126,488	21	128,159	1,222,450
Equivalent period, 1931	32	27	5	219,750	24	190,800	2,429,875
Total, 1931	72	52	20	450,960	58	417,327	4,455,595
					Fire-tube Boilers		
April, 1932	..	..	..	.....	..	.....	.....
April, 1931	1	..	1	4,000	2	6,700	45,000
January to April (inclusive, 1932)	4	..	4	4,800	4	8,000	41,000
Equivalent period, 1931	12	1	11	15,000	12	20,204	143,200
Total, 1931	35	11	24	39,300	37	59,761	347,100

REVIEW OF NEW TECHNICAL B

Any of the books reviewed on this page may be secured from In-Ce-Co Publishing Corporation, 200 Madison Avenue, New York

The Book of Metals

By Donald Wilhelm

THIS book is typical of several others published in recent years, notably those by Sir James Jeams on the science of the universe, in that it discloses the principles and practices of a thoroughly technical industry in such form that it is just as easily assimilated by the layman as by the technician. The story of metals is one that will engage the interest of nearly everyone, for who is not interested in the origin, the process of manufacture and the use of iron, steel and various other metals which are the principal ingredients of industrial development as we know it today and which have been dominant factors in civilization and the progress of man for several thousand years past?

Mr. Wilhelm writes of all this in a style that is as entertaining as it is informative. He takes up each metal in turn, describes the sources of supply, tells how it is taken from the ground, refined and manufactured. In the chapters devoted to ferrous metals, he takes the reader into the great steel mills and gives him an intimate picture of the amazing processes through which the metal passes to emerge in the various forms required for its manifold uses to man.

This book should be read by everyone who is interested in knowing the whys and wherefores of the principal metals, and even by those who are perhaps fairly well informed in these matters but who would like to have a glimpse into the great steel laboratories of today so that they might know more of the innumerable alloys which are being developed there to meet the future needs of industry.

Book contains 332 pages, size 6½ x 9½, price $4.00.

General Engineering Handbook

(First Edition)

Editor-in-Chief: Charles Edward O'Rourke

THIS is a handbook for all engineers. The civil engineer will find eleven sections of this compact handbook dealing with his branch alone; he will also find the mechanical engineering data he needs in nine other sections. He will find five sections that give him the electrical engineering information he needs.

The plan for this handbook, according to the editor-in-chief, who is Assistant Professor of Structural Engineering at Cornell University, was conceived in the belief that a great amount of fundamental engineering data could be assembled in a compact pocketbook of not over 900 pages, which would be valuable as a ready reference and hand companion for practicing engineers and for students. The purpose is to supply a compact reference work of important fundamentals for all engineers.

The author seems to have accomplished this in a most satisfactory manner. There are 31 sections, each section dealing with one general subject. Six of these sections contain material which is of importance to all engineers; others belong in the fields of civil, mechanical, electrical engineering, etc. Also engineering mathematics.

The book contains tables and charts useful in the various branches of engineering. An important factor is a bibliography of the important works dealing with the subject, at the end of each chapter. Also a comprehensive index.

The book is bound handsomely in red fabrikoid, size 5 x 8, contains 922 pages, price $4.00.

Power Plant Management

(Job Analysis and Functions)

By Walter N. Polakov

THIS book offers information of considerable value to those concerned with the management or supervision of power plants, and will help to solve many problems.

Today power plant cost comes next to pay-roll as an expense item in textile finishing, paper products, food industries and other enterprises. In the application of efficient methods—checking layouts, costs, waste, etc.—in the generation, distribution and utilization of power, and especially in the scientific organization of the human element, savings of tens and hundreds of thousands of dollars are possible. It is to show management's place in creating these savings—to outline the simple proceedings necessary for the typical power plant, large or small—that this book was planned.

The author, a consulting engineer with over 20 years' management experience, in power plant practice, has drawn on this authoritative background and embodies in his book material which is of great value to the power plant manager and probably not obtainable in any other medium.

The book contains 171 pages, size 5½ x 7½; cloth binding. Price $2.00.

BOOKS FOR THE ENGINEER

1—Draft and Capacity of Chimneys

By J. G. Mingle

339 pages Illustrated Price $3.50

The most authoritative book ever published on the subject of draft and chimneys. The subject matter has been developed primarily from a theoretical standpoint and then amplified by experimental data gleaned from actual practice.

The author observes that draft, even to the expert, frequently contains an element of mystery, and that there is often a great deal of confusion and misconception on the subject. A careful study of this book will give a thorough and practical knowledge of this important subject.

The book is profusely illustrated with graphs and charts. Many valuable tables are included and the index has been prepared for ready reference.

2—A Handbook of English in Engineering Use

By A. C. Howell

308 pages Price $2.50

Here is a real up-to-the-minute handbook that should be on the desk of every technical writer. Most engineers have occasion to do considerable writing and will find this useful. Chapters are devoted to word usage and idioms, sentence and paragraph structure, composition, punctuation and the mechanics of writing and grammar. Examples cover letters, reports and technical articles.

3—Steam Tables and Mollier Diagram

By Joseph H. Keenan

Price $2.00

These new Steam Tables, extending to a pressure of 3,500 lb. per sq. in. and a temperature of 1,000 deg. fahr., were developed from the latest experimental data secured by investigators in laboratories of Europe and those of the United States. The Symbols used in this work are taken from the latest test prepared by the A. S. A. Sub-Committee for Heat and Thermodynamics. A large copy of the new Mollier Diagram (23" x 34") is also included.

4—Mechanical Engineers' Hand-Book

By Lionel S. Marks

2265 pages Price $7.00

The third edition of this great reference book has been thoroughly revised in all parts, bringing it up to date in both practice and theory. Important new sections include vibration problems—refractories, high temperature, carbonization of coal and gas making—low-temperature carbonization of coal—industrial combustion furnaces—electric industrial furnaces.

5—Handbook of Oil Burning

By Harry F. Tapp

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